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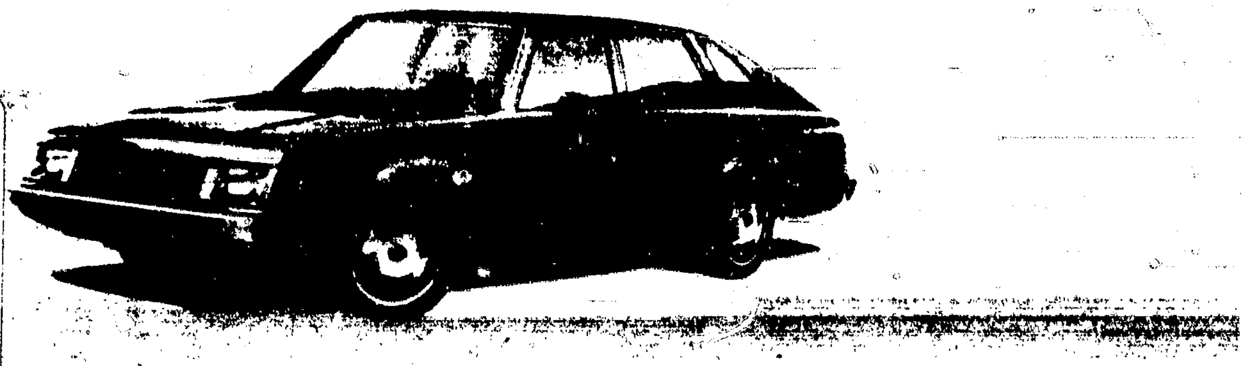
NEAR-TERM HYBRID VEHICLE PROGRAM

FINAL REPORT — PHASE I

Appendix B — Design Trade-Off Studies Report

Volume I — Design Trade-Off Studies



Contract No. 955190

Submitted to

Jet Propulsion Laboratory
California Institute of Technology
4800 Oak Grove Drive
Pasadena, California 91103

Submitted by

General Electric Company
Corporate Research and Development
Schenectady, New York 12301

October 8, 1979

GENERAL ELECTRIC

SRD-79-134/3

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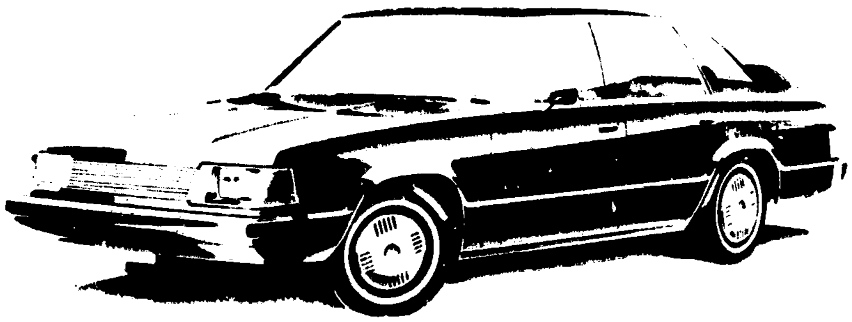

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FOREWORD

The Electric and Hybrid Vehicle (EHV) Program was established in DOE in response to the Electric and Hybrid Vehicle Research, Development, and Demonstration Act of 1976. Responsibility for the EHV Program resides in the Office of Electric and Hybrid Vehicle Systems of DOE. The Near-Term Hybrid Vehicle (NTHV) Program is an element of the EHV Program. DOE has assigned procurement and management responsibility for the Near-Term Hybrid Vehicle Program to the California Institute of Technology, Jet Propulsion Laboratory (JPL).

The overall objective of the DOE EHV Program is to promote the development of electric and hybrid vehicle technologies and to demonstrate the validity of these systems as transportation options which are less dependent on petroleum resources.

As part of the NTHV Program, General Electric and its subcontractors have completed studies leading to the Preliminary Design of a hybrid passenger vehicle which is projected to reduce petroleum consumption in the near term (commencing in 1985). This work has been done under JPL Contract 955190, Modification 3, Phase I of the Near-Term Vehicle Program.

This volume is part of the Deliverable Item 7, Final Report of the Phase I studies. In accordance with Data Requirement Description 7 of the Contract, the following documents are submitted as appendices:

APPENDIX A is the Mission Analysis and Performance Specification Studies Report that constitutes Deliverable Item 7 and reports on the works of Task 1.

APPENDIX B is a three-volume set that constitutes Deliverable Item 2 and reports on the work of Task 2. The three volumes are:

- Volume I --Design Trade-Off Studies Report
- Volume II --Supplement to Design Trade-off Studies Report, Volume I
- Volume III --Computer Program Listings

APPENDIX C is the Preliminary Design Data Package that constitutes Deliverable Item 3 and reports on the work of Task 3.

APPENDIX D is the Sensitivity Analysis Report that constitutes Deliverable Item 8 and reports on Task 4.

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The three classifications - Appendix, Deliverable Item, and Task number - may be used interchangeably in these documents. The interrelationship is tabulated below:

Appendix	Deliverable Item	Task	Title
A	1	1	Mission Analysis and Performance Specification Studies Report
B	2	2	Vol. I - Design Trade-Off Studies Report
			Vol. II - Supplement to Design Trade-Off Studies
			Vol. III - Computer Program Listing
C	3	3	Preliminary Design Data Package
D	8	4	Sensitivity Analysis Report

This is Volume I, Design Trade-Off Studies Report of Appendix B. It presents the study methodology, component characterization, power train configuration classification and definition of terms, evaluation and comparison of candidate power trains, vehicle design analysis and layout trade-off, control strategy trade-off, component selection and sizing, guidelines for the preliminary design task, and summary and major findings.

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Section 1

INTRODUCTION AND SUMMARY

Section 1

INTRODUCTION

1.1 INTRODUCTION

This is Volume I, Design Trade-Off Studies Report of Appendix B. It reports on the work of Task 2 and is part of Deliverable Item 7, Final Report, which is the summary report of a series which document the results of Phase I of the Near-Term Hybrid Vehicle Program. This phase of the program was a study leading to the preliminary design of a five-passenger hybrid vehicle utilizing two energy sources (electricity and gasoline/diesel fuel) to minimize petroleum usage on a fleet basis.

The program is sponsored by the U.S. Department of Energy (DOE) and the California Institute of Technology, Jet Propulsion Laboratory (JPL). Responsibility for this program at DOE resides in the Office of Electric and Hybrid Vehicle Systems. Work on this Phase I portion of the program was done by General Electric Corporate Research and Development and its subcontractors under JPL Contract 955190.

This volume presents the study methodology; evaluation and comparison of candidate power trains; control strategy; and the selected design concept.

1.2 OBJECTIVES OF THE DESIGN TRADE-OFF STUDIES (TASK 2)

The objective of Task 2 of the Design Trade-off Studies was to select a design concept for the Near-Term Hybrid Vehicle which offers the greatest promise for achieving the program objective of maximizing the potential for reducing petroleum consumption.

At the same time, the concept selected must meet or exceed the constraints and minimum requirements given in Exhibit I of the RFP and the vehicle performance specifications defined in Appendix A, Mission Analysis and Performance Specification Studies Report.

1.3 SUMMARY

A three-step approach was used in conducting the Design Trade-off Studies. First, individual drive-line components were analyzed for specific weight and cost, and vehicle synthesis calculations were made to evaluate various hybrid/electric power train configurations. In the second step, second-by-second simulations of the most promising hybrid power train identified in the first step were made on the computer for urban and highway driving cycles to compare in detail the candidate configurations and to determine the component sizes needed to satisfy vehicle mission requirements. The third step involved consideration of the trade-offs required to package, in a five-passenger vehicle, the most promising power train configurations simulated in detail. The methodology followed in this study is discussed in detail in Section 2.

Section 3, "Component Characterization," of this Design Trade-off Studies Report presents gross component characteristics such as specific weight and specific cost, as well as detailed operating characteristics of each component in the power train. Subcontractor reports and memos on the components are included in Volume II. The following power train components are discussed:

- Heat engines
- Electric motors and controllers
- Batteries
- Transmissions and power combination units
- Microprocessors

Various hybrid power train trade-offs are discussed in Section 5, "Evaluation and Comparison of Candidate Power Trains." The conclusions reached in Section 5 were based on vehicle synthesis calculations made using a computer program HYVELD (Hybrid Vehicle Design) which was prepared for that purpose. The major design trade-offs considered were:

- Parallel versus series configurations
- Secondary energy storage
- Heat engine/electric drive power split
- Battery type
- Heat engine type
- Electric drive type

Each trade-off was examined for a range of vehicle power-to-weight ratios and design electric range values. Primary criteria for vehicle comparison were:

- Total weight
- Selling price
- Ownership cost
- Fuel saving
- Dollar saving

Vehicle design and layout trade-offs are discussed in Section 6. Various approaches to packaging the hybrid power train in a five-passenger vehicle were studied and evaluated. Factors considered in the evaluation were vehicle handling and safety, passenger comfort, and power train maintenance. Special attention was also given to accessory requirements and how they could be provided in the hybrid/electric vehicle. As a result of the design and layout studies, vehicle and power train configurations were identified for more detailed consideration in the Preliminary Design Task.

The control strategy for operating the electric and heat engine drive systems is critical in meeting the design goals of the hybrid/electric vehicle. Various aspects of the development of the control strategy are discussed in Section 7. The control strategies were tested using a second-by-second hybrid vehicle simulation program HYVEC (Hybrid Vehicle Calculation). The following factors were examined:

- Primary drive system selection and power sharing
- Shift logic
- Regenerative braking
- Battery switching
- Accessories

The effect of these factors and drive-line component characteristics on vehicle fuel economy, total energy use, emissions, and acceleration performance were studied using the HYVEC program. The results of the hybrid vehicle simulations are presented in Section 8, "Component Selection and Siting Trade-offs for Various Driving Cycles." Guidelines for the Preliminary Design Task and the performance and energy use characteristics of the vehicle are summarized in Section 9.

1.3.1 MAJOR FINDINGS

The major findings from the Design Trade-Off Studies are:

1. The parallel configuration with a 60/40 split between peak power of the heat engine and electric drive systems was near optimum from the standpoints of vehicle weight, ownership cost, and energy usage (fuel and electricity).
2. Based primarily on economic considerations, a dc electric drive system utilizing a separately excited motor with field control and battery switching was selected for the Near-Term Hybrid Vehicle.
3. The prime heat engine candidates are a fuel-injected gasoline engine and a turbocharged diesel. Both engines are 1.6 ℓ in displacement and develop about 70 hp. The diesel engine yielded 25 to 30% better fuel economy in the hybrid application than the gasoline engine, but technology does not currently exist to reduce the NO_x and particulate emissions of the diesel to levels considered acceptable by the Environmental Protection Agency for 1985. The diesel also has possible cold-starting problems when used in an on/off mode.

4. A complex control strategy involving integrated power sharing between the heat engine and the electric drive systems is required for the hybrid vehicle to have acceleration performance equivalent to a conventional ICE vehicle and, at the same time, high fuel economy and acceptable electric range. Implementation of the control strategy developed in the computer simulations will require the use of microprocessors in the hybrid vehicle control system.

5. The hybrid vehicle simulations showed that 700 lbs of ISOA lead-acid batteries yielded satisfactory electric range and vehicle acceleration performance. The Ni-Zn batteries were found to be the most attractive for the hybrid application, but there is considerable uncertainty concerning the cycle lifetime and cost of Ni-Zn batteries in the 1982 to 1985 time period.

6. The vehicle layout studies showed that the complete hybrid power train including the lead-acid batteries could be packaged in the engine compartment of the 1979 Chevrolet Malibu without any intrusion into the passenger compartment.

7. The initial selling price (in 1978 dollars) of the hybrid vehicle was calculated to be about $7000 compared with $5700 for a conventional ICE vehicle of the same performance and passenger-carrying capacity. The ownership (life cycle) cost of the hybrid was calculated to be 17.8 ¢/mi compared with 18.5 ¢/mi for the Reference Vehicle for energy costs of $1.00/gal for gasoline and 4.2 ¢/kWh for electricity. The lifetime of the hybrid vehicle was taken to be 12 yrs compared with 10 yrs for the conventional ICE vehicle.

8. Detailed hybrid vehicle simulations showed that for the first 30 mi (the electric range of the vehicle) in urban driving, the fuel economy was 80 mpg using a gasoline engine and 100 mpg using a diesel engine. Over the first 75 mi, the average fuel economy of the hybrid was 42 mpg for the gasoline engine and 55 mpg using the diesel engine. The highway fuel economy of the hybrid vehicle is slightly better than that of the Reference ICE Vehicle. In urban driving the hybrid would save about 75% of the fuel used by the conventional vehicle, and in combined urban/highway driving the fuel saving is about 50%.

1.3.2 VEHICLE PERFORMANCE AND ENERGY-USE CHARACTERISTICS

The performance and energy-use characteristics of the hybrid vehicle (gasoline engine-powered) found to be near-optimum in the design trade-off studies are given in Tables 1-1 and 1-2. The preliminary design of the vehicle is the goal of Task 3 of the Near-Term Hybrid Vehicle Program.

Table 1-1
VEHICLE PERFORMANCE SPECIFICATIONS

P1	Nonrefueled Range	
P1.1	FHDC (Gasoline - 10 gal Tank)	550 km
P1.2	FUDC*	120 km, > 425 km
P1.3	J227a(B) (Electricity Only)	80 km
P2	Cruise Speed	130 km/h
P3	Maximum Speed	
P3.1	Maximum Speed	150 km/h
P3.2	Length of Time Maximum Speed Can Be Maintained on Level Road	1 min
P4	Accelerations	
P4.1	0-50 km/h (0-30 mph)	4.0 s
P4.2	0-90 km/h (0-56 mph)	11.0 s
P4.3	40-90 km/h (25-56 mph)	7.8 s

P5 Gradability

	Grade	Speed	Distance
P5.1	3%	100 km/h	km (Unlimited)†
P5.2	5%	95 km/h	km (Unlimited)
P5.3	8%	80 km/h	km (Unlimited)
P5.4	15%	40 km/h	km (Unlimited)
P5.5 Maximum Grade	25%	—	—

P6	Payload Capacity (Incl. Passengers)	535 kg
P7	Cargo Capacity	0.5 m^3
P8	Consumer Costs	
P8.1	Consumer Purchase Price (1978 $)	$7000
P8.2	Consumer Life Cycle Cost (1978 $)	0.11 $/km
P9	Emissions - Federal Test Procedure** (Gasoline Engine)	
P9.1	Hydrocarbons (HC)	0.06 gm/km, 0.12 gm/km
P9.2	Carbon Monoxide (CO)	0.40 gm/km, 0.75 gm/km
P9.3	Nitrogen Oxides (NO_x)	0.40 gm/km, 0.64 gm/km

*First number is the range at which the batteries must be recharged from the heat engine; second number is range at which the 40 liter gasoline tank is empty.
†On heat engine alone
**The first number corresponds to first 50 mi, second to 120 mi

Table 1-2

ENERGY CONSUMPTION MEASURES

E1	Annual petroleum fuel energy consumption per vehicle compared to reference vehicle over contractor-developed mission[a]	30,000 MJ SAVED[b]
E2	Annual total energy consumption[c] per vehicle compared to reference vehicle over contractor-developed mission[a]	6,000 MJ SAVED[b]
E3	Potential annual fleet petroleum fuel energy savings compared to reference vehicle over contractor-developed mission[c]	30×10^9 MJ
E4	Potential annual fleet total energy consumption[c] compared to reference vehicle over contractor-developed mission[d]	6×10^9 MJ SAVED[b]
E5	Average energy consumption[c] over maximum nonrefueled range	
	E5.1 FHDC (gasoline only)	2.53 MJ/km (32 mpg)
	E5.2 FUDC[e]	2.86 MJ/km, 3.1 MJ/km, 3.6 MJ/km
	E5.3 J227a (B) (electricity only)	2.45 MJ/km
E6	Average petroleum fuel energy consumption over maximum nonrefueled range	
	E6.1 FHDC	2.53 MJ/km (32 mpg
	E6.2 FUDC[e]	1.01 MJ/km (80 mpg), 1.93 MJ/km (42 mpg), 3.2 MJ/km (25 mpg)
	E6.3 J227a (B)	0 MJ
E7	Total energy consumed[c] versus distance traveled starting with full charge and full tank over the following cycles	
	E7.1 FHDC	2.53 MJ/km (Not a Function of Distance)
	E7.2 FUDC	(See Figure 8-8)
	E7.3 J227a (B)	2.45 MJ/km (Not a Function of Distance)
E8	Petroleum fuel energy consumed versus distance traveled starting with full charge and full tank over the following cycles[f]	
	E8.1 FHDC	2.53 MJ/km (Not a Function of Distance)
	E8.2 FUDC	(See Figure 8-1)
	E8.3 J227a (B)	0 MJ/km (Not a Function of Distance)

1 MJ = 0.278 kWh = 948 Btu = .00758 gal gasoline

10^9 MJ/yr = 452 barrels crude oil/day

(a) Mission is 11,852 mi/yr; 65% EPA urban cycle, 35% EPA highway cycle

(b) The annual fuel and energy usages of the Reference ICE Vehicle (1985 model) are 456 gallons of gasoline and 60,158 MJ. A fleet of one million Reference Vehicles would use 60×10^9 MJ.

(c) Includes energy needed to generate the electricity at the power plant (35% efficiency)

(d) For one million hybrid vehicles replacing one million Reference Vehicles

(e) The first number corresponds to the first 50 km; the second number to 120 km; the third number to 425 km, at which the gasoline tank is empty

(f) Does not include petroleum consumption resulting from generation of wall plug electricity used by the vehicle

Section 2

STUDY METHODOLOGY

Section 2

STUDY METHODOLOGY

2.1 GENERAL APPROACH

The approach used in the Design Trade-off Studies consisted of several steps. The first step involved the synthesis of total vehicle weight and cost from the specific weights and costs of individual components for several candidate configurations. In this initial screening of components and drive-line configurations, the component and vehicle energy-use characteristics were averaged over the driving cycles of interest. In this first step, a wide range of drive-line components and combinations were considered using a Hybrid Vehicle Design Program (HYVELD) for the computer calculations. The objective of the vehicle-level screening was to identify those drive-line components and arrangements which are most attractive for more detailed consideration in the next step of the screening procedure.

The second step of the trade-off study involved second-by-second simulation of the hybrid/electric vehicle designs operating over several driving cycles. This simulation required detailed modeling of the various drive-line components and the control strategy for operation of the electric and heat engine drive systems. In this second step, vehicle characteristics, such as drag coefficient, frontal area, weight, etc., were fixed. The major emphasis was to determine the effect on electricity and gasoline use of power train changes, such as battery type and weight, engine type, motor voltage control technique, and variations in control strategy. The second-by-second vehicle simulations were performed using the computer program HYVEC (Hybrid Vehicle Calculations).

The third step in the Design Trade-off Study was to determine if the attractive hybrid power train arrangements could be packaged in a five-passenger car and if so, what were the primary considerations in comparing one power train layout to another.

2.2 POWER TRAIN COMPONENTS AND CONFIGURATIONS CONSIDERED

There are a myriad of possible hybrid/electric power train configurations and components which could be considered in design trade-off studies. Hence, some technical judgment was used at the outset of the study to reduce the contenders to manageable proportions. For instance, the following generic hybrid arrangements were considered and then excluded:

- Electric drive through individual wheel-mounted motors
- The split power train in which one set of wheels is driven by the heat engine and the second set by the electric motor

Wheel-mounted motors were excluded because it was felt that for the passenger car size vehicles such motors are collectively less efficient, heavier, and more expensive than a single motor of the same combined horsepower. The split power train arrangement was ruled out because the control of such a system when there is power sharing between the heat engine and electric drives would present great difficulty with respect to flexibility and smoothness. In addition, the split power train arrangement is inherently heavier and more expensive than single drive shaft configurations.

The hybrid power train configurations and components considered in the present trade-off studies are listed in Table 2-1. As indicated in the table, both series and parallel configurations were analyzed in the first screening step, and a number of candidate components were studied for each function in the drive line. The effect of vehicle range and power-to-weight ratio on the relative attractiveness of the various component candidates from both the vehicle weight and cost points-of-view were investigated using the HYVELD computer program.

2.3 COMPONENT CHARACTERIZATION

In order to perform the trade-off studies it was necessary to characterize each of the components in Table 2-1. The degree of detail required for each component depended on whether it was included only in the vehicle level (first step) screening or in both the vehicle level and second-by-second simulation screenings. For the initial screening, each component was characterized in terms of specific weight (lb/kW) and specific cost ($/kW). For the second-by-second simulations, detailed characterization of the components was required including efficiencies (and/or losses) over the complete operating range (power and speed) of the component. For the batteries it was necessary to obtain charge/discharge characteristics over a wide range of charge/discharge currents. For the most part, the components were characterized using data taken on existing hardware.

2.4 VEHICLE AND POWER TRAIN SPECIFICATIONS

In order to synthesize the power train, it is necessary to specify a number of vehicle characteristics and the degree of power sharing between the heat engine and electric drive systems. For the hybrid vehicle design calculations using HYVELD, the vehicle characteristics required are baseline chassis weight, payload, energy consumption per ton-mi, fraction of the energy from heat engine, and the performance parameters - power-to-weight ratio and range on electricity. The power sharing between the heat engine and electric drive systems is specified in terms of the fraction of the peak power attainable from each drive system. The efficiency of the drive-line is specified as a single value averaged over the driving cycles of interest. As noted previously, the effect of the vehicle and power train specifications on the attractiveness of the various components is of particular importance.

Table 2-1

HYBRID POWER TRAIN CONFIGURATIONS AND COMPONENTS CONSIDERED IN THE DESIGN TRADE-OFF STUDY

General Power Train Arrangements

1. Series
2. Parallel

Heat Engines

1. Fuel-injected Gasoline (N.A.)
2. Diesel (N.A. and T.C.)
3. Uniform Charge Rotary
4. Single-Shaft Gas Turbine
5. Stirling

Transmissions/Clutches

1. Power Addition with Differential Action
2. Multi-Speed Shifted Gearbox with Clutch
3. Torque Converter with Lock-up
4. Continuously Variable (CVT)

Electric Drives

1. DC Separately Excited with Armature and/or Field Control
2. AC Induction with Pulse-Width Modulated Inverter

Batteries (Primary Storage)

1. Lead-Acid
2. Ni-Zn
3. Ni-Fe
4. LiAl-FeSx

Secondary Storage

1. Flywheel
2. Lead-Acid Batteries

2.5 METHODOLOGY FOR THE EVALUATION AND COMPARISON OF CANDIDATE POWER TRAINS

During the initial screening of the candidate hybrid/electric power trains, comparisons were made in terms of total vehicle weight, initial and operating costs, break-even gasoline price, and total energy used. These comparisons were made for fixed baseline vehicle chassis weight and vehicle performance specifications. The vehicles utilizing hybrid/electric power trains were also compared with the reference Internal Combustion Engine (ICE) passenger car and an all-electric car having similar utility to a car owner. For all of these comparisons, economic factors such as interest rate, discount rate, finance period, payback period, inflation rate, etc. were held constant. In addition, the fuel economy of the reference ICE car was fixed. Complete lists of the design and economic factors which were varied or held constant in the initial screening study are given in Table 2-2.

Candidate power trains included in the second-by-second simulation studies were compared in terms of range primarily on battery-stored electricity, fuel economy (mpg), heat engine emissions, and energy use. These comparisons were made for urban/suburban, highway and intra-city driving using appropriate combinations of the Environmental Protection Agency's urban and highway cycles and the SAE J227a Schedule B cycle. In addition, the 0-60 mps and 40-60 mph acceleration times obtained for the various candidate hybrid power trains were compared.

2.6 VEHICLE-LEVEL POWER TRAIN LAYOUT CONSIDERATIONS

The results of the design trade-off studies yielded the power ratings of the heat engine and electric drive systems and the weight of the batteries needed to meet the vehicle performance and range requirements set forth by the Mission Analysis (Task 1). In addition, the trade-off studies identified particular components, such as heat engines, electric motors, and batteries, which are prime candidates for use in the Preliminary Design (Task 3). In order to investigate various options for packaging power train components of the required size into a five-passenger car, preliminary vehicle layouts were made using the 1979 Chevrolet Malibu (chassis and interior seating arrangement) as the baseline design. Various placements of the motor, engine, and batteries were made including front-and-rear-wheel drive and fore-and-aft-positioning of the batteries. These layouts formed the basis for trade-off considerations involving crash worthiness, handling, vehicle weight, and ease of battery maintenance.

2.7 CONTROL STRATEGY AND VEHICLE OPERATION ON VARIOUS DRIVING CYCLES

Selection and evaluation of power train components must include careful consideration of the control strategy to be used. The control strategy involves coordinating use of the heat engine and electric drive systems. The power and speed requirements of the vehicle

Table 2-2

VEHICLE AND ECONOMIC FACTOR INPUT PARAMETERS FOR THE DESIGN TRADE-OFF CALCULATIONS

Hybrid/Electric Design Parameter	
Baseline Chassis Weight	*
Payload Weight	*
Power-to-Weight Ratio	
Range (Design) - All-Electric	
Range (Design) - Hybrid	
Electric Drive-Line Efficiency	
Cost of Additional Chassis Weight	*
Weight Propagation Factor	*
Miles Traveled per Year	*
Fraction of Miles in City	*
Energy Consumption in City (kWh/ton-mi)	*
Energy Consumption on Highway (kWh/ton-mi)	*
Fraction of Energy from Engine in City	*
Fraction of Energy from Engine in Highway	*
Price of Electricity	*
Specific Cost of Motor/Generator ($/kW)	
Specific Cost of Generator ($/kW)	
Specific Cost of Controller ($/kW)	
Specific Weight of Motor/Generator ($/lb)	
Specific Weight of Generator ($/lb)	
Specific Weight of Controller ($/lb)	
Average Engine bsfc in City	*
Average Engine bsfc on Highway	*
Time for Sustained Power from the Flywheel	*
Conventional Vehicle Design Parameters	
Power-to-Weight Ratio	
Specific Weight of Engine	
Specific Weight of Transmission	
Specific Cost of Engine	
Specific Cost of Transmission	
Fuel Economy in City	*
Fuel Economy on Highway	*
Consumer Cost	*
Price of Gasoline	*
Maintenance Cost per Mile	*
Economic Factors	
Discount Rate	*
Inflation Rate	*
Interest Rate	*
Payback Period	*
Finance Period	*
Tax Rate	*
Sales Tax	*

*Input Parameters Held Constant in Vehicle Synthesis Calculations must be matched to the capabilities of the engine and motor. Power matching is accomplished by means of a transmission and/or power combination differential. The control strategy should be self-adaptive to varying levels of battery charge and rates of acceleration and deceleration. In addition, the control parameters for the various components should be easily sensed and used as inputs to the system controller. All of these aspects of developing and implementing a control strategy for the efficient, flexible, and smooth operation of the hybrid/electric power train were considered in the trade-off studies and will be referred to frequently throughout the discussions of Sections 3, 5, 7, and 8.

Section 3

COMPONENT CHARACTERIZATION

Section 3

COMPONENT CHARACTERIZATION

3.1 INTRODUCTION

As noted in Section 2, the vehicle design computer program (HYVELD) requires as inputs gross component characteristics, such as specific weight (lb/kW) and specific cost ($/kW and $/kWh). The vehicle simulation program (HYVEC) requires a detailed knowledge of the operating characteristics of each of the components in the power train. Both types of characteristics are given in this section for the following power train components:

- Heat engines
- Electric motors and controllers
- Batteries
- Transmissions and power combination units
- Microprocessors

Considerable information/data was developed in Task 2 for each of the components by General Electric Corporate Research and Development (GE CRD) and its subcontractors. The subcontractor reports and memos produced during the study for each of the components are included in the supplement to this report. These documents served as references for this section and are cited frequently. This section deals primarily with the results of the component studies and the characteristics used in the vehicle trade-off and simulation studies to be treated in later sections. The characterization discussion of each component has the following format:

1. Types considered
2. Gross characteristics of all types
3. Operating characteristics for those types included in the simulation studies

3.2 HEAT ENGINES

3.2.1 TYPE OF ENGINES CONSIDERED

As indicated in Table 3-1, there are many heat engines which can be considered for hybrid vehicles ranging from the conventional gasoline and diesel engines to the more advanced gas turbine and Stirling engines. All of these engines have been reviewed and compared in a recent JPL publication.(1) In the current study, emphasis was placed on obtaining detailed information on small fuel-injected gasoline engines (for example, the Volkswagen 1.6 ℓ). The results of the work done by the General Electric

Table 3-1

HEAT ENGINES CONSIDERED FOR HYBRID VEHICLE

Type	Example	Status	Advantages	Disadvantages
Gasoline (Reciprocating)				
Carbureted	VW 1.6ℓ	Production	Highly Developed and Readily Available	Good But Not Excellent Fuel Economy
Fuel Injected	VW 1.6ℓ	Production		
Three-Valve CVCC	Honda 1.6ℓ	Production		
Gasoline (Rotary)				
Uniform Charge	Mazda RX-7, Two-Rotor, 100 hp	Production	Small Size and Weight	Relatively High-Emissions and Lower Fuel Economy Than Reciprocating Engine
Stratified Charge	Curtis Wright	Lab Testing	Improved Fuel Efficiency Over Uniform Change	Uncertain Status of Development
Diesel				
Naturally Aspirated	VW 1.5ℓ	Production	Excellent Fuel Efficiency	High Weight and Size, Noise and NO_x Emissions
Turbocharged	VW 1.5ℓ	Pre-Production Vehicle Testing	Reduced Size and Weight Compared with N.A.	Response of Turbocharger, Difficulty in Controlling NO_x, Uncertainty of Unregulated Emissions
Stirling	United Stirling P40	Lab Testing and Limited Vehicle Testing	Potential Excellent Fuel Efficiency	High Weight and Size, Uncertain Status of Development, High Cost
Gas Turbine				
Two-Shaft	Chrysler Upgraded (7th gen), 104 hp	Lab Testing	Relatively Small Size	Low Fuel Economy, High Cost, Uncertain Status of Development
Single-Shaft	Williams Research WR 34, 50 hp	Prototype for Tractor	Small Size and Weight	Not Developed as an Automotive Engine, High Cost

Space Systems Division in that area are included in the heat engine supplement in Volume II of this report. In addition, information on the WR34 single-shaft gas turbine was obtained from Williams Research as that was the only known gas turbine engine in the required horsepower range for the hybrid vehicle application. Material on the WR34 engine is also included in the heat engine supplement.

A summary of the status and the advantages/disadvantages of the various engines in their present state-of-development is given in Table 3-1. The advantages/disadvantages will be considered in quantitative terms in the next section in which the weight, size and fuel economy and emissions, and cost characteristics of the engines are discussed.

3.2.2 SPECIFIC CHARACTERISTICS

3.2.2.1 Weight and Size

In the hybrid vehicle application, the weight and size of the engine are very important because such vehicles tend to be considerably heavier and require more drive-line components than the conventional ICE vehicle with which they are compared. Hence, engines which are small and light are particularly attractive for use in hybrid vehicles. Heat engine size and weight characteristics are shown in Table 3-2 in terms of specific weight (lb/kW) and specific volume (ft^3/kW). The engines are rated at their peak output. It is convenient to group the engines into small, medium, and large categories for later design trade-off comparisons. The specific weight and volume of the three groups are shown in Table 3-3.

3.2.2.2 Fuel Economy and Emissions

Detailed comparisons and projections of the fuel economy and emissions of passenger cars using various types of conventional and advanced engines are given in Reference 1. Results taken from this source are shown in Figures 3-1 and 3-2. These comparisons pertain to the Environmental Protection Agency driving cycles and vehicles utilizing engines in the conventional manner. For passenger car applications, the various engines can be ranked as to their attractiveness from the fuel economy and emissions points-of-view. This is done in Table 3-4 for both the present and future potential capabilities of the engines. Only the Stirling engine shows high potential in both fuel economy and emissions, and it is currently in a very early stage of development.

It is clear from Table 3-4 that a trade-off between fuel economy and emissions is almost unavoidable at the present time in selecting the "best" engine for a conventional ICE passenger car. Clearly, at this time the choice is between the gasoline and diesel engines. This same choice is encountered for the hybrid vehicle. From a fuel economy standpoint in conventional cars, the diesel engine has two clear advantages:

1 Superior part-load fuel consumption (lb/bhp-hr)

2. Much lower idle fuel flow

Table 3-2

HEAT ENGINE SIZE AND WEIGHT CHARACTERISTICS(1)

Type	lb/kW	ft^3/kW
Gasoline (Reciprocating)		
Carbureted	5.0	0.08
Fuel-Injected	5.0	0.08
Three-Valve CVCC	7.0	0.10
Gasoline (Rotary)		
Uniform Charge	4.0	0.07
Stratified Charge	4.2	0.07
Diesel		
Naturally Aspirated	7.8	0.12
Turbocharged	5.6	0.09
Stirling	7.5	0.12
Gas Turbine		
Two-Shaft	5.1	0.10
Single-Shaft	4.4	0.09

Table 3-3

CATEGORIZATION OF HEAT ENGINES BY SPECIFIC WEIGHT AND VOLUME

Category	Examples	lb/kW	ft^3/kW
Small	Gasoline-Rotary Single-shaft GT	4.0	0.07
Medium	Gasoline-Reciprocating Turboch-Diesel Two-shaft GT	5.5	0.08
Large	Naturally Aspirated Diesel Stirling	7.6	0.12

These advantages are shown graphically in Figures 3-3 and 3-4. Another advantage of the diesel engine which is not usually exploited in passenger cars is that the fuel can be cut off much of the time during deceleration.

All of these advantages of the diesel engine will be reduced in the hybrid vehicle application. First, it is very likely that

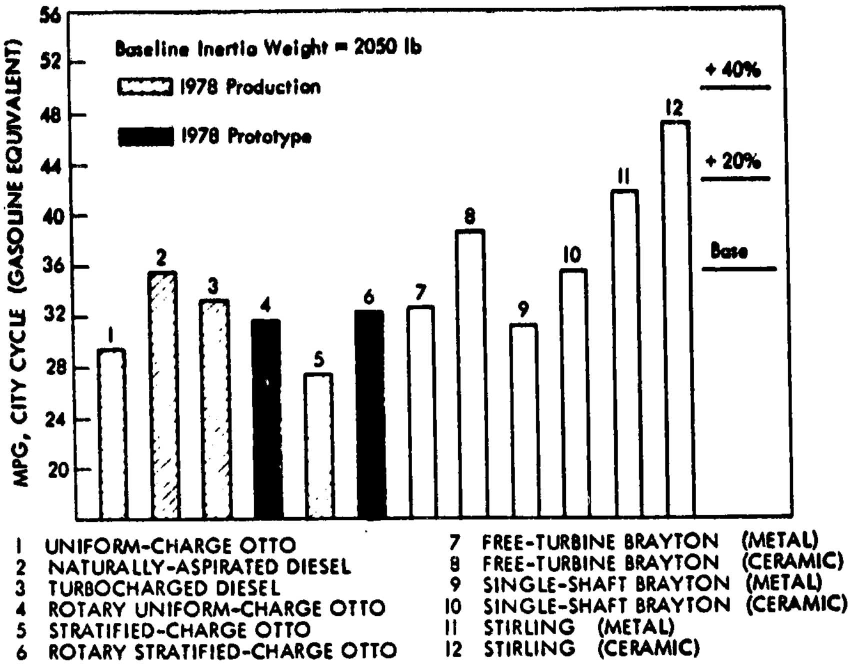

Figure 3-1. City Fuel Economy Comparison for Small Otto-Engine Equivalent Vehicles (1.0 g/mi NO_x)

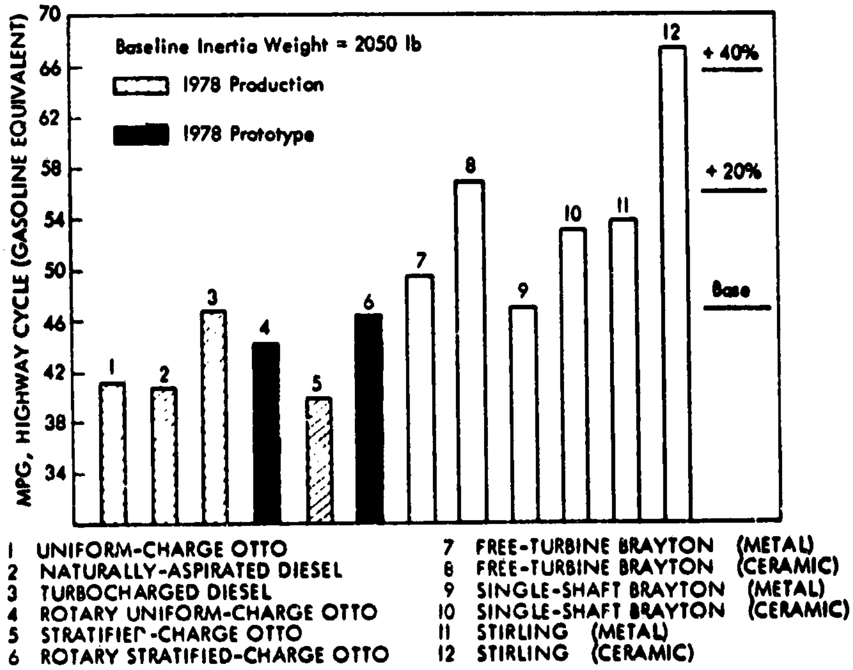

Figure 3-2. Highway Fuel Economy Comparison for Small Otto-Engine Equivalent Vehicles (1.0 g/mi NO_x)

Table 3-4

FUEL ECONOMY AND EMISSIONS RANKING

Fuel Economy Ranking

Present	Potential
Diesel	Stirling
Gasoline-Receprocating	Diesel
Gasoline-Rotary	Gasoline-Reciprocating
Stirling	Gasoline-Rotary
Gas Turbine*	Gas Turbine*

Emissions Ranking

Present	Potential
Gasoline (Three-way Catalyst)	Stirling
Stirling	Gasoline (Three-way Catalyst)
Gas Turbine	Gas Turbine
Diesel†	Diesel†

*Engine sizes less than 100 hp
†Based on NO_x, particulates, odor

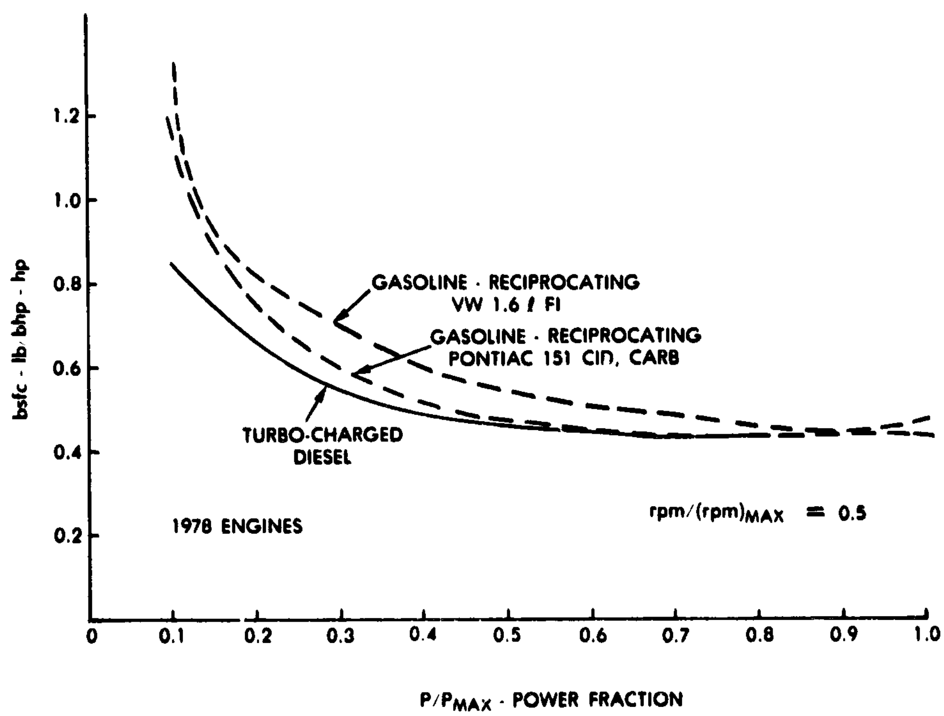

Figure 3-3. Comparisons Between 1978 Diesel and Gasoline Engines

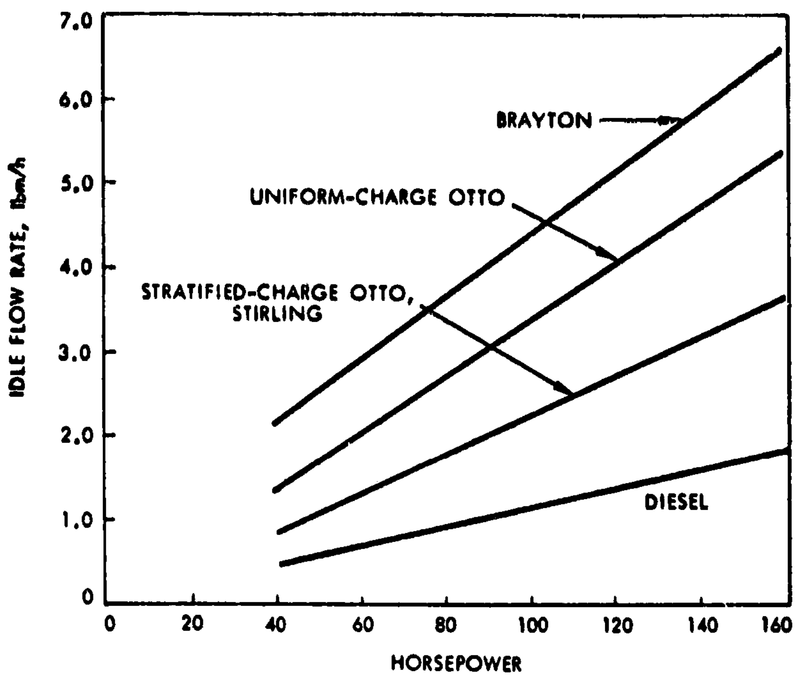

Figure 3-4. Idle Fuel Flow Rate Characteristics

the heat engine will be used in an on-off mode and, thus, the engine would never idle. Secondly, the engine will be smaller (lower peak horsepower for the same size vehicle) and so will seldom operate at part-loads less than 25 to 30% maximum power at any revolutions per minute. This will be true on the highway as well as in urban driving. Thirdly, if a fuel-injected gasoline engine is used, its fuel flow would be cut off during most decelerations. Figure 3-3 indicated that at load fractions greater than about 30%, the differences between the brake specific fuel consumption diesel and a gasoline engine tailored for good part-load efficiency can be quite small. Hence, in terms of fuel economy, the diesel does not seem to have any inherent advantage in the hybrid application.

The next consideration is the comparison of emissions for gasoline and diesel engines. In terms of carbon monoxide and hydrocarbon emissions, the pre-chamber diesel is inherently much cleaner than the gasoline engine without exhaust gas treatment. The diesel engine also has somewhat lower NO_x emissions than the gasoline engine without Exhaust Gas Recovery (EGR). With the use of catalysts (oxidation and three-way), the exhaust emissions from the gasoline engine can be greatly reduced to meet the most stringent emission standards of 0.4 gm/mi hydrocarbon, 3.4 gm/mi of carbon monoxide, and 0.4 gm/mi of oxides of nitrogen. Unfortunately, reducing the NO_x emissions from the diesel engine is much more difficult than for a gasoline engine. A catalyst cannot be used in the exhaust because the diesel operates lean. The use of EGR is problematical due to the particulates in the diesel exhausts. Hence, as far as the regulated pollutants (HC, CO, NO_x)

are concerned, the diesel is relatively clean without emission control, but a further reduction of NO_x emissions to meet future standards will be difficult. In the case of the gasoline engine, the untreated emissions are relatively high, but all three regulated pollutants can be reduced using catalysts to meet present and future emission standards.

In addition to the difficulty in reducing NO_x emissions, the diesel engine has a tendency to smoke at high loads and even at lower loads has micron-size carbon particles in the exhaust stream. Current diesel-powered cars have a particulate emission of 0.5 to 1.0 gm/mi. Diesel exhaust also exhibits an odor which is objectionable. The odor problem is worst at idle and other light-load conditions. Therefore, diesel odor would be much less of a problem for hybrid vehicles than for conventional ICE cars. There is considerable uncertainty at the present time concerning the so-called unregulated emissions of the diesel, but it does not seem wise to ignore them in light of the probability of future Federal Standards in those areas.

The choice between using a gasoline or a diesel engine in the Hybrid vehicle will be a difficult one and must await the results of the second-by-second vehicle simulation discussed in Section 8. Detailed characteristics of both types of engines have been developed and are discussed later in this section.

As noted in Section 3.2.2.1, low engine weight and size is particularly attractive in the hybrid vehicle application. In this regard, the gasoline-rotary engine should be considered. As indicated in Table 3-2, the rotary gasoline engine is somewhat lighter and smaller than the reciprocating gasoline engine. If a rotary engine in the 60 to 80-hp range were available in production and had emission and fuel economy characteristics comparable to a gasoline reciprocating engine, the rotary engine would be a prime candidate for use in the hybrid vehicle. The only production rotary engine at the present time is the two-rotor, 100-hp engine used by Mazda in the RX-7. The horsepower of that engine is too large for efficient use in the five-passenger hybrid vehicle. In addition, little work has been done to adapt the three-way catalyst system to the rotary engine. Recent publications (References 4,5) indicate that considerable progress is being made by several groups in improving both the fuel economy and emission characteristics of rotary engines. Also, there is speculation in the industry that Mazda plans to produce in the next several years, a single-rotor engine of the 60 to 80-hp class for use in sub-compact cars. Such an engine would be quite attractive for the hybrid vehicle if it could be used with a three-way catalyst and had fuel consumption characteristics comparable to the conventional gasoline engine. However, the size/weight advantage of the rotary is not sufficiently great that the hybrid application demands its use. The size and weight of a four-cylinder engine (high-speed gasoline or turbocharged diesel) can be easily accommodated in the hybrid vehicle. Also, the engineering and development cost of using a conventional four-cylinder engine in the hybrid would undoubtedly be much less than the cost of using a less well-developed rotary engine.

Table 3-5

THE OEM COST OF CONVENTIONAL GASOLINE ENGINES

Model	Engine	Horsepower	1975 OEM Cost	1978* OEM Cost	($/kW) (Without Catalyst)	($/kW)† (With Catalyst)
Chevelle	6 cyl, 250 CID	100	331	405	5.4	6.8
Audi	4 cyl, 90 CID	75	340	416	7.3	9.1
Rabbit	4 cyl, 90 CID	70	291	356	6.8	8.5
Pinto	4 cyl, 122 CID	90	322	395	5.9	7.4

*A 7% annual inflation rate was assumed for the period 1975 to 1978.
†Cost increase of 25% is assumed for addition of catalyst for emission control.

3.2.2.3 Engine Costs

There has been considerable uncertainty concerning the cost of conventional gasoline engines and even greater uncertainty concerning the cost of other types of engines. Recent studies of automotive component costs by Pioneer Engineering and Manufacturing Company(6) have made available a data base which is at least internally consistent. A summary of the Pioneer results for the OEM cost of a number of conventional four- and 6-cylinder gasoline engines is given in Table 3-5. The resultant specific engine costs for the engines without a catalyst are 5.5 to 7.5 \$/kW. It has been assumed that the cost would increase by 25% with the addition of a catalyst for emissions control. In that case, the range of the specific cost is 6.5 to 9.0 \$/kW.

Estimation of the costs of engine types other than gasoline reciprocating are more difficult because published information on those costs is not based on a detailed breakdown of the engines part-by-part as was done in the Pioneer Study for the conventional gasoline engines. The approach taken in this study to estimate the cost of the other engine types is to use the cost factors (the ratio of the cost of each engine type to that of the gasoline reciprocating engine) given in Reference 1. The cost factors used are given in Table 3-6 along with the resultant specific costs for the various types of engines. Except for the Stirling and gas turbine engines, the projected cost differences are relatively small (10 to 20%).

3.2.3 DETAILED ENGINE CHARACTERISTICS

The following detailed engine characteristics are needed to perform the second-by-second simulation of the hybrid vehicle over various driving cycles using HYVEC:

- A fuel consumption map
- An emissions (HC, CO, NO_x) map
- Motoring losses
- Rotating inertia

In addition, modeling of the warmup and cool-down of the catalyst in the engine's emission control system is needed. All the above detailed engine characteristics have been developed for the fuel-injected four-cylinder gasoline engine and the turbocharged four-cylinder diesel since they are the prime candidates for use in the hybrid vehicle. Fuel consumption map information for all other engine types is given in Reference 1 in the tabular form required for input into the vehicle simulation program (HYVEC). Those engines have not been utilized in the present study because they were not considered prime candidates.

Table 3-6

SPECIFIC COSTS (OEM) FOR VARIOUS TYPES OF HEAT ENGINES

Engine Type	Cost Factor	Specific Cost ($/kW) OEM
Gasoline with Catalyst	1.0	8.5
Diesel		
Naturally Aspirated	1.1	9.4
Turbocharged	1.2	10.2
Stirling	1.6	13.6
Gas Turbine		
Two-Shaft	1.45	12.3
One-Shaft	1.40	11.9

Considerable information on four-cylinder, fuel-injected gasoline engines is given in the heat engine supplement (Volume II of this report). Much additional information on the Volkswagen (VW) 1.6-ℓ engine was obtained from the United States Department of Transportation, Transportation Research Center.(7) In total, all the required data for the steady-state operation of the VW 1.6-ℓ engine was obtained for developing inputs for the HYVEC simulation program. Steady-state fuel consumption (bsfc-lb/bhp-hr) and emission (HC, CO, NO_x-gm/bhp-hr) maps are given in Tables 3-7 and 3-8. The emissions correspond to a point upstream of the catalyst. In the vehicle simulation program, the emission rate for each pollutant is reduced according to an assumed conversion efficiency for that pollutant. The effect of catalyst warmup and cool-down was included as follows:

$$\frac{EF}{EFO} = 1-e^{-ETON/TCHW}$$

$$\frac{TCHW}{TCHWO} = 1-e^{-ETOF/TCCHC}$$

where EF ≡ catalyst conversion efficiency

EFO ≡ catalyst conversion efficiency for warmed-up catalyst

Table 3-7

BRAKE SPECIFIC FUEL CONSUMPTION
FOR A VOLKSWAGEN 1.6-ℓ EFI ENGINE

Idle fuel flow = 1.1 kg/hr

Idle rpm = 900

hp_{MAX} = 72 hp = 53.7 kW

WE_{MAX} = 6000

	% Power								
% Speed	10	14	29	43	57	72	86	100	Limit Power Curve P_{MAX} (%)
15.0	3.06	2.150	1.400	1.030	0.800	0.720	0.650	0.650	11.1
20.0	1.98	1.600	1.051	0.753	0.617	0.566	0.526	0.521	19.4
26.6	1.76	1.500	0.866	0.660	0.539	0.513	0.486	0.476	26.3
33.3	1.39	1.120	0.743	0.579	0.521	0.495	0.474	0.446	38.2
40.0	1.29	1.190	0.727	0.595	0.513	0.496	0.461	0.447	45.8
47.0	1.20	1.112	0.715	0.582	0.513	0.495	0.463	0.445	53.6
53.0	1.20	1.110	0.720	0.588	0.533	0.493	0.472	0.452	61.1
60.0	1.29	1.194	0.750	0.597	0.521	0.479	0.467	0.447	69.0
67.0	1.33	1.219	0.742	0.605	0.525	0.492	0.467	0.452	76.8
76.7	1.29	1.179	0.765	0.608	0.550	0.507	0.485	0.468	88.1
86.7	1.36	1.242	0.789	0.630	0.566	0.526	0.505	0.487	100.0
100.0	1.70	1.500	0.900	0.746	0.642	0.590	0.554	0.528	100.0

Table 3-8a

HYDROCARBON EMISSIONS FOR A VW 1.6-ℓ EFI ENGINE (gm/bhp-hr)

Idle HC = 10 gm/hr

Idle rpm = 900

		% Power							
% Speed	RPM	10	14	29	43	57	72	86	100
15.0	780	8.00	6.00	4.30	3.70	3.70	4.68	4.29	5.00
20.0	1200	3.10	3.00	2.42	2.13	1.97	1.85	1.82	2.10
26.6	1600	3.40	3.25	2.66	2.32	2.03	1.79	1.70	1.75
33.3	2000	3.55	3.33	2.65	2.31	2.26	1.98	1.94	1.81
40.0	2400	3.60	3.40	2.73	2.56	2.37	2.05	1.93	1.94
47.0	2800	3.60	3.40	2.68	2.49	2.18	1.98	1.75	1.99
53.0	3200	3.65	3.48	2.81	2.47	2.27	2.13	1.95	1.77
60.0	3600	3.50	3.30	2.62	2.19	2.11	1.89	1.72	1.43
67.0	4000	3.40	3.24	2.52	2.34	2.81	1.85	1.66	1.52
76.7	4600	3.10	2.93	2.42	2.08	1.99	1.77	1.62	1.48
86.7	5200	3.00	2.79	2.24	2.02	1.91	1.67	1.53	1.37
100.0	6000	3.02	2.92	2.58	2.30	2.05	1.78	1.54	1.38

Table 3-8b

CARBON MONOXIDE EMISSIONS FOR A VW 1.6-ℓ
EFI ENGINE (gm/bhp-hr)

Idle CO = 100 gm/hr

Idle rpm = 900

		% Power							
% Speed	RPM	10	14	29	43	57	72	86	100
13.0	900	120.0	85.0	43.0	37.0	35.0	32.0	87.0	150.0
20.0	1200	60.0	49.0	27.0	19.0	16.6	15.5	17.3	29.8
26.6	1600	47.0	40.5	27.0	19.0	17.6	17.1	16.9	17.6
33.3	2000	35.0	32.8	23.4	17.7	18.0	18.9	19.7	16.8
40.0	2400	43.0	37.0	24.0	22.7	20.7	20.7	17.3	16.7
47.0	2800	40.0	36.4	26.7	23.1	20.4	18.0	16.2	15.3
53.0	3200	46.0	41.0	29.6	24.5	20.3	17.5	15.8	14.9
60.0	3600	55.0	46.9	32.5	23.5	19.3	17.4	15.8	15.9
67.0	4000	54.0	47.7	28.8	23.2	17.9	16.6	16.2	15.4
76.7	4600	51.0	45.5	27.0	20.3	18.2	17.2	15.7	14.4
86.7	5200	45.0	38.7	26.4	20.2	18.2	17.0	15.7	14.7
100.0	6000	74.0	67.0	46.7	30.8	21.6	19.0	17.6	15.9

Table 3-8c

NITROGEN OXIDES EMISSIONS FOR A VW 1.6-ℓ
EFI ENGINE (gm/bhp-hr)

Idle NO_x = 2 gm/hr

		% Power							
% Speed	RPM	10	14	29	43	57	72	86	100
15.0	900	2.20	2.30	2.50	2.7	3.20	5.50	6.50	8.00
20.0	1200	2.55	2.60	2.80	3.3	4.20	6.75	7.35	8.64
26.6	1600	2.75	2.90	3.75	5.1	6.35	7.25	8.20	9.60
33.3	2000	2.80	4.20	6.40	8.9	11.40	11.10	12.10	12.90
40.0	2400	4.50	6.20	8.50	11.1	13.40	13.20	14.10	14.90
47.0	2800	6.60	7.10	9.30	11.6	14.10	13.50	14.50	15.50
53.0	3200	9.00	9.50	11.70	14.2	16.40	15.50	16.20	17.30
60.0	3600	9.20	9.83	12.80	15.7	17.00	16.00	16.90	17.50
67.0	4000	13.40	13.90	15.70	18.3	19.30	18.00	17.80	18.70
76.7	4600	17.80	19.30	21.70	22.4	22.80	20.30	21.10	22.10
86.7	5200	21.80	22.40	23.80	24.8	25.30	22.50	22.60	23.00
100.0	6000	16.00	17.50	22.30	25.0	24.80	23.20	23.00	23.50

ETON ≡ engine time on

TCHW ≡ catalyst warmup time

TCHWO ≡ catalyst warmup time from a completely cold condition

TCCHC ≡ time required for essentially complete cool-down of the catalyst

ETOF ≡ engine time off

It has been assumed in the vehicle simulation studies that a three-way catalyst would be used. In addition, it appears likely that the catalyst will utilize a metallic rather than a ceramic substrate because the metallic substrate catalyst exhibits a shorter warmup time and can withstand higher temperatures without loss of activity. Metallic substrate catalysts are discussed in some detail in References 8 and 9. For the computer simulations, the following values were used for the catalyst characteristics:

$$EFO = \begin{cases} HC & - 80\% \\ CO & - 90\% \\ NO_x & - 80\% \end{cases}$$

$$TCHWO = 20 \text{ s}$$

$$TCCHC = 120 \text{ s}$$

In the hybrid vehicle application, the heat engine will be used much of the time in an on/off operating mode. Hence, there are times when the engine will be in a motoring condition without fuel injection. The losses associated with engine motoring were studied, and the effect of valve deactivation on those losses was considered. Valve deactivation involves keeping the inlet and exhaust valves closed unless fuel is being injected. In this way, the pumping losses associated with moving air through the engine are avoided and the motoring losses reduced. Various means of implementing valve deactivation are discussed in References 10 and 11, as well as in the heat engine supplement. Motoring losses with and without valve deactivation are shown in Figure 3-5 for the VW 1.6-ℓ engine. Even with valve deactivation the motoring losses are quite large for engine speeds greater than 50% of $(rpm)_{max}$. This means that very careful consideration must be given to when and how the heat engine is brought up to speed prior to its activation.

Computer simulations of hybrid power trains utilizing a turbocharged diesel engine are discussed in Section 8. Of prime interest is the fuel economy penalty incurred by using a gasoline engine rather than the diesel engine and the NO_x and particulate emissions using the diesel. The fuel consumption map (bsfc) used for the diesel engine is given in Table 3-9 and Figure 3-6. This map was obtained from DOT/TSC, Cambridge and is for the turbocharged version of the VW diesel Rabbit engine (90CID) which has a peak power rating of 70 hp at 5500 rpm.

Emissions data for diesel engines from Reference 12 was used to determine the emission maps for hydrocarbons, carbon monoxide, and NO_x given in Tables 3-10 through 3-13. After discussions with Volkswagen, West Germany on how to convert Bosch Number readings for exhaust smoke to particulate concentrations, the map given in Table 3-13 was calculated. The data given in Tables 3-10 through 3-13 were used to characterize the turbocharged diesel engine in the second-by-second hybrid vehicle simulation studies.

Table 3-9

BRAKE SPECIFIC FUEL CONSUMPTION FOR A TURBOCHARGED DIESEL ENGINE (lb/bhp-hr)

hp_{MAX} = 70 hp
WE_{MAX} = 5000
Idle Fuel Flow = 0.545 kg/hr

	% Power								
% rpm	10	20	30	40	50	60	80	100	% $Power_{MAX}$
20	1.080	0.850	0.690	0.580	0.540	0.515	0.500	0.525	16.8
30	0.970	0.740	0.600	0.515	0.470	0.465	0.455	0.490	28.8
40	0.840	0.660	0.540	0.485	0.465	0.450	0.445	0.480	47.6
50	0.840	0.660	0.540	0.485	0.465	0.450	0.445	0.480	63.0
60	0.840	0.660	0.540	0.485	0.465	0.450	0.445	0.480	76.6
80	0.950	0.760	0.640	0.565	0.510	0.485	0.465	0.490	94.6
90	1.075	0.885	0.740	0.640	0.583	0.540	0.495	0.500	98.0
100	1.150	1.000	0.835	0.725	0.650	0.605	0.565	0.570	100.0

Table 3-10

HYDROCARBON EMISSIONS FOR A TURBOCHARGED DIESEL ENGINE (gm/bhp-hr)

	% Power							
% Speed	10	20	30	40	50	60	80	100
20	0.50	0.38	0.300	0.24	0.210	0.200	0.17	0.14
30	0.50	0.38	0.300	0.22	0.195	0.175	0.15	0.10
40	0.50	0.38	0.300	0.22	0.175	0.145	0.10	0.08
50	0.53	0.41	0.340	0.26	0.190	0.140	0.10	0.08
60	0.53	0.45	0.410	0.37	0.30	0.220	0.13	0.08
80	0.45	0.62	0.870	1.08	1.08	0.860	0.45	0.20
90	0.23	0.50	0.600	0.65	0.690	0.710	0.61	0.29
100	0.26	0.29	0.315	0.34	0.35	0.370	0.35	0.26

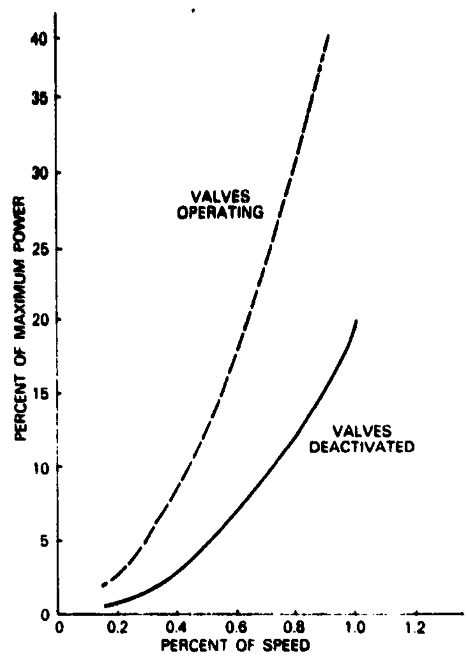

Figure 3-5. VW Engine Motoring Losses

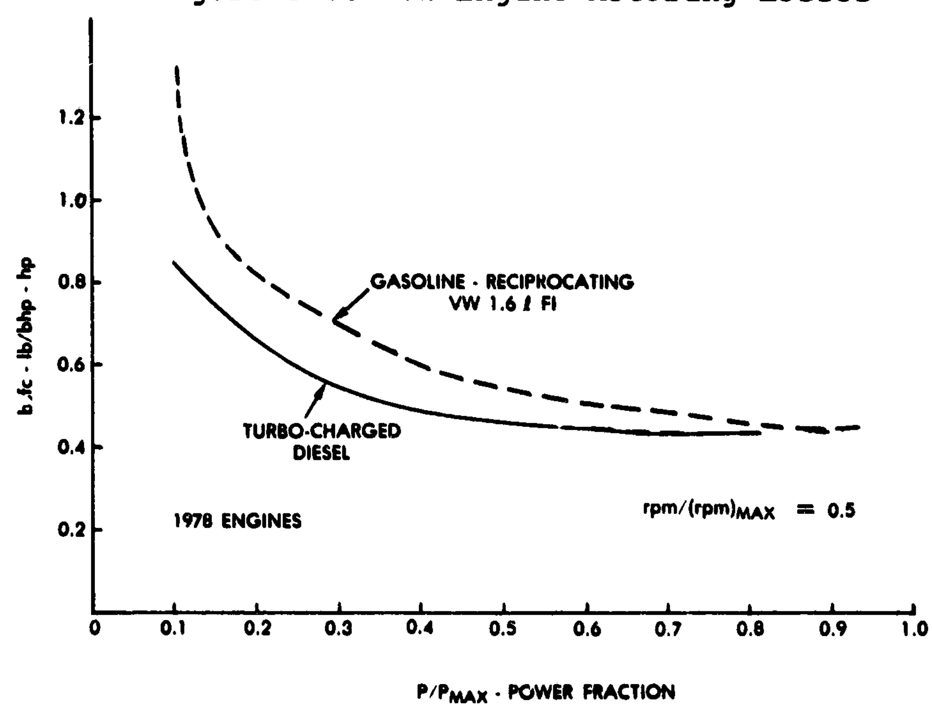

Figure 3-6. Comparison of Gasoline and Diesel Engine Fuel Economy

Table 3-11

CARBON MONOXIDE EMISSIONS FOR A TURBOCHARGED DIESEL ENGINE
(gm/bhp-hr)

% Speed	% Power 10	20	30	40	50	60	80	100
20	4.00	3.20	2.60	2.30	2.30	2.80	4.40	6.70
30	3.70	2.80	2.10	1.60	1.50	1.50	2.10	6.70
40	3.10	2.20	1.60	1.20	1.10	1.10	1.10	1.90
50	3.50	2.60	1.90	1.40	1.20	1.10	1.10	1.50
60	4.50	3.40	2.40	2.00	1.90	1.60	1.10	1.30
80	0.50	0.55	0.80	1.00	1.00	1.00	0.50	0.25
90	0.25	0.30	0.45	0.60	0.70	0.75	0.60	0.30
100	0.55	0.55	0.55	0.55	0.50	0.37	0.37	0.26

Table 3-12

NITROGEN OXIDES EMISSIONS FOR A TURBOCHARGED DIESEL ENGINE
(gm/bhp-hr)

% Speed	% Power 10	20	30	40	50	60	80	100
20	5.7	4.4	3.7	3.0	2.6	2.4	1.8	1.2
30	5.7	4.4	3.7	3.0	2.6	2.4	2.0	1.5
40	5.7	4.4	3.7	3.0	2.6	2.4	2.0	1.5
50	6.2	5.2	4.2	3.4	3.0	2.7	2.3	1.9
60	7.3	6.2	4.7	3.9	3.4	3.1	2.7	2.2
80	10.0	8.8	8.1	6.6	5.9	5.3	4.0	2.8
90	10.3	9.1	8.4	7.3	6.8	6.2	4.8	3.1
100	10.4	9.0	7.7	6.8	6.2	5.6	4.5	3.1

Table 3-13

CARBON SOOT EMISSIONS FOR A TURBOCHARGED DIESEL ENGINE
(gm/bhp-hr)

% Speed	% Power 10	20	30	40	50	60	80	100
20	2.24	1.92	1.73	1.73	1.92	2.17	2.92	4.1
30	.98	.66	.464	.40	.43	.59	1.30	1.83
40	.96	.66	.46	.40	.41	.44	.60	1.81
50	.85	.60	.44	.39	.38	.36	.34	.53
60	.96	.74	.61	.55	.56	.57	.53	.67
80	1.79	1.67	1.58	1.40	1.31	1.11	.73	.80
90	1.99	1.84	1.68	1.56	1.43	1.21	.74	.59
100	2.28	2.18	1.76	1.36	1.01	.82	.53	.38

3.3 ELECTRIC MOTORS AND CONTROLLERS

3.3.1 TYPES OF ELECTRIC DRIVE SYSTEMS CONSIDERED

The components considered for use in the electric drive system of the hybrid vehicle are essentially the same as those being developed for use in all-electric vehicles. There has been much activity in recent years to advance the state-of-the-art of motors and controllers for use in electric vehicles. The new motor technology has been assessed for use in the hybrid vehicle. As will be noted in later sections, the power rating of the electric drive system needed for the the five-passenger parallel hybrid vehicle is essentially the same as that of the smaller four-passenger electric vehicle for which most of the advanced electric drive systems are being developed. This has considerably simplified the task of comparing various electric drive approaches because the comparisons can be made based on actual designs and, in some cases, actual hardware of the desired power rating.

As indicated in Figure 3-7, there are a number of electric motors which can be considered for use in the electric drive system of the hybrid vehicle. The motors fall into two broad categories: dc and ac. The power and control characteristics of the motor determine the type of power conditioning equipment (i.e., the controller) which must be utilized between the battery and the motor to permit the operator to drive the vehicle. At the present time, most electric vehicles use dc motors with the series type being commercially available in relatively large volume (50,000/yr for forklift truck and golf-cart applications). For the higher speed automotive applications, dc separately excited motors have been found to be more advantageous (move efficient, lower battery and motor current, the possibility of field control and elimination of armature control) than dc series motors, but that type of dc

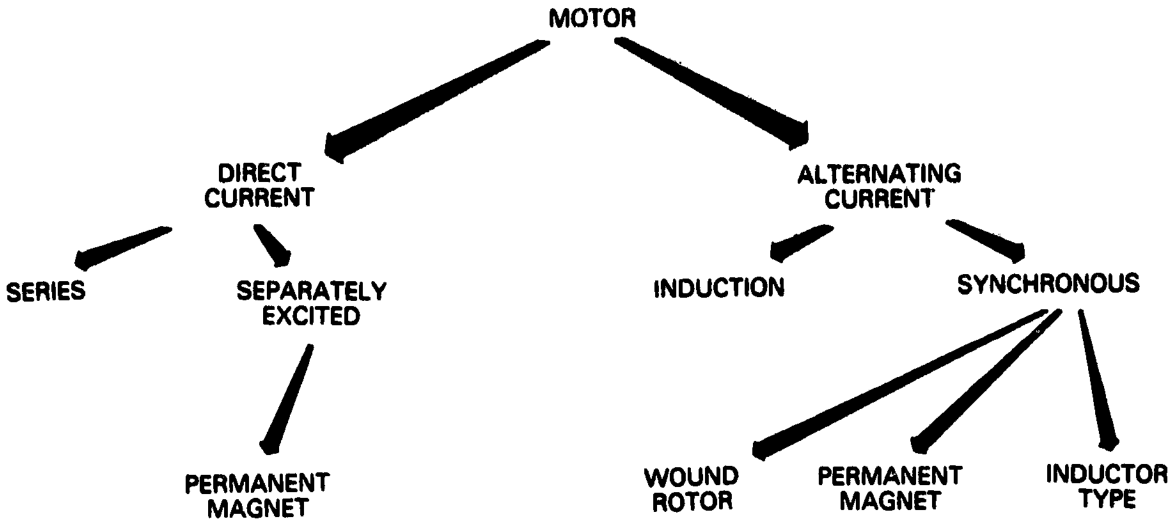

Figure 3-7. Candidate Motor Types

motor is not yet available in large quantities. Most in-use electric vehicles utilize a chopper-type controller to modulate the armature current to the motor and thus the power of the electric motor.

At the present time, most chopper controllers use SCR thyristor devices in the controller circuits to switch on and off the current to the motor from the battery. Recent developments in power transistor technology are making it possible to use transistors in place of the SCR thyristors with a resultant reduction in weight, size, complexity, and cost of the controller. The electric drive system developed by General Electric for use in the Near-Term Electric Vehicle for the Department of Energy and the Jet Propulsion Laboratory utilizes power transistors and a microprocessor-controlled dc separately excited motor. Evolutionary development of that drive system is taken in the present study to be the state-of-the-art of the dc drive system which can be expected to be available for use in the Near-Term Hybrid Vehicle Program. Hence, the characteristics of that drive system are used as the baseline with which to compare other candidate electric drive systems.

Alternating current (ac) drive systems have also been considered for use in hybrid/electric vehicles. The primary attractiveness of the ac system is that the ac motor is smaller and lighter than the dc motor of the same power rating and maximum rpm, and the advantage of the ac motor can be widened markedly by increasing its maximum rpm. This is possible with the ac motor because it has electronic rather than mechanical commutation. In the dc motor at high rpm, retention of the commutator bars and brush bounce and wear become problems. In addition, ac motors are produced in large volume for household and industrial applications, and it is likely their cost will be lower than for dc motors. The major disadvantage of the ac drive system is that it requires a complex and expensive power conditioning device (inverter) between the dc power source (the battery) and the ac motor. Since the motor requires poly-phase (normally three-phase), alternating current to operate, the inverter represents essentially three dc chopper controllers in weight, size, complexity, and cost. Up to the present time, the ac controller requirements have precluded the use of ac drive systems in electric vehicles intended for commercialization. With the advent of power transistors and the development of higher speed ac motors, there is increased activity at General Electric and elsewhere to re-evaluate the possible use of ac drive systems in hybrid/electric vehicles. The results of these recent developments and reassessments at General Electric are included in the present study. Considerable detail concerning General Electric work in the ac drive area is given in the electric drive supplement (Volume II) attached to this report.

A summary of the status and the advantages/disadvantages of the various electric drive systems shown in Figure 3-7 is given in Table 3-14. The advantages/disadvantages will be considered in quantitative terms in the next section in which the size and weight, efficiency and loss, and cost characteristics of most of the electric drive types are discussed.

3.3.2 SPECIFIC CHARACTERISTICS

3.3.2.1 Weight and Size

The weight and size characteristics of the various electric drive system components are shown in Table 3-15. It should be noted that it is customary to rate electric motors in terms of their power output in continuous operation (1 to 2 hr) and not in terms of the peak power that can be sustained for only 1 to 2 min or less due to heating effects. Since heat engines are rated in terms of peak power, the specific power density (lb/kW) values of the electric motor should be divided by a factor of two before comparison with the corresponding heat engine values. Power conditioning units (controllers) are rated in terms of peak power because the switching devices (thyristors and power transistors) are rated by maximum current which for a given voltage limit corresponds to peak power.

First consider the weight and size characteristics of the dc drive systems. As indicated in Table 3-15, dc motors tend to be heavy because they operate at relatively low speed and are mechanically commutated. In terms of continuous-rated specific power, there is not much difference between series and separately excited dc motors, but for automotive applications in which it is important to be able to use the power at high vehicle speeds (also high motor rpm), the series motor has a lower usable power capability because the torque falls off more rapidly at increased rpm due to system voltage limitations. In the case of separately excited dc motors

Table 3-14

ELECTRIC DRIVE MOTORS CONSIDERED FOR HYBRID VEHICLES

Type	Example	Status	Advantages	Disadvantages
dc Series	GE BT2366	Production	No Separate Field Excitation Required, High Torque at Low Speeds, Easy to Control	Large Torque Decrease at High Speeds Is Difficult to Control. Regeneration Requires Reversal of Field Windings
dc Separately Excited	Modified Version of GE BT2366 for Use in DOE/SPL NT Electric Car Available Also from Bosch, Siemens, and Lucas.	Limited Production	Control Flexibility, Ease of Achieving Regenerative Braking	Relative Complexity of Construction, Max rpm Limited by Commutation
ac Induction (Wound Rotor and Stator)	GE Test Motor in NASA Contract No. DEN.3-59, Max rpm - 12,000	Laboratory Testing	Small Size and Light Weight, Lower Cost	Requirement for Complex Three-Phase Inverter
ac Permanent Magnet Synchronous Disk Motor	GE Test Motor on CRD Funding	Laboratory Testing	Small Size and Lightweight, High-Efficiency Load-Commutated	Difficult to Start and Control Due to Very Low Reactance, Requires Three-Phase Inverter, Cost Uncertain

Table 3-15

WEIGHT AND SIZE CHARACTERISTICS OF ELECTRIC DRIVE SYSTEM COMPONENTS

Drive Type	Motor* lb/kW	Motor* ft^3/kW	Controller† lb/kW
dc - Series Motor	14.2	0.075	1.2 (1.9)**
dc - Separately Excited Motor	12.3	0.075	1.2 (1.9)
ac Induction Motor			
- low speed (5000 rpm)	7.7		2.4 (3.9)
- high speed (12,000 rpm)	5.6	0.045	2.4 (3.9)
ac PM Synchronous Disc Motor	5.6		4.8 (3.5)

*Motor is rated at continous (1 hr) power (kW)
†Controller is rated at peak power (kW)
**A(B); A - weight using transistors, B - weight using thyristors

the field can be weakened at higher rpm, and it is not necessary to limit armature current as severely. This is one of the primary reasons that series motors are not preferred for automotive applications. It can be expected that traction dc motors will become lighter (lower lb/kW) as motors are designed to meet specific auto cycle applications. Direct current motors having specific power values of 10 lb/kW or less are likely. It is apparent from Table 3-15 that for the dc drive system, the weight of the controller is a relatively small fraction of the system weight using either thyristors or transistors. This fraction will, however, increase as the motor weights are reduced. In the vehicle synthesis calculations discussed in Section 5, specific weights of 10 lb/kW and 1.5 to 2 lb/kW were used for the dc motor and controller electronics, respectively. These values are consistent with the expected trend that motor weights will decrease, and the complexity, and thus the weight of the power conditioning and microprocessor package, will increase in the next several years.

Next consider the ac drive systems. It is clear from Table 3-15 that the specific weights of the ac motors are significantly less than those of the dc motors. This is particularly true of the high-speed induction and PM synchronous disc motors. Both of these motors have specific power density values less than the heat engine on a comparable peak power rating. As noted previously, the power conditioning units (controller) for the ac motors are considerably heavier and more complex than those needed by the dc motors. For the ac induction motors which need electronically forced commutation, an inverter using thyristors would weigh more than one-half that of the motor. Using power transistors, the controller weight fraction is reduced to one-half to one-third of the motor, and the total weight of the motor and controller is significantly less than that of the dc system. An ac drive system using the high-speed induction motor and a transistorized pulse width modulated (PWM) inverter is currently being built by General Electric under contract to NASA-Lewis. The power torque capability of that drive package is comparable to the dc drive package being used in the DOE/GE Near-Term Electric Car. It weighs about 100 lbs less than the dc unit. Present studies indicate that an ac drive system could be built to meet the requirements of the hybrid vehicle application, but as discussed in a later section, the OEM cost of the ac drive unit would be considerably higher than that of the lowest cost dc drive unit. This will be discussed in more detail later.

Development of the PM synchronous disc motor is in a much earlier stage than that of the ac induction motor. Preliminary laboratory tests of the motor are currently in progress. The synchronous motor is load-(or self-) commutated which means that it is not necessary to provide circuitry in the inverter to turn off and on the semiconductor switching devices (thyristors or transistors). The voltage requirement of the PM motor is about twice that of the induction motor because no field weakening is possible at the higher motor rpm. Thyristors are available which can withstand the higher voltages, but transistors are not. Hence, thyristors are currently the best choice for use in the inverter for the PM motor. Using thyristors, the weight of the PM motor plus inverter would be significantly heavier than that of the induction motor system. The PM motor is more efficient than the induction motor and, with advances in transistor technology, could lead to a lighter unit. Starting and control of the PM synchronous motor is, however, more difficult than for the induction motor, and those problems would have to be solved. Because of the relatively early stage of development of the ac PM motor, it is not considered further in this study, and only the ac inductor motor will be included in the vehicle synthesis and simulation tasks.

3.3.2.2 Efficiency and Losses

The efficiency of the electric drive-line depends on the efficiency of the motor and controller and the power required by the various electronic control devices and the motor cooling fan. It is advantageous to identify the various losses as their relative importance changes with motor speed and output torque.

Motor Losses

- Windage
- Bearings
- I^2R losses in windings
- Core magnetic losses
- Commutation (dc)
- Cooling

Controller (Inverter) Losses

- Switching losses
- Switching device conduction loss (IΔE)
- Commutation (ac)
- Capacitor losses
- Cooling

The efficiencies of the motor and controller at relatively high load are given in Table 3-16. The decrease in efficiency at part load is illustrated in Figures 3-8, 3-9. Even though the efficiencies of the various electrical components are relatively high, it is apparent that it is difficult to assign an average efficiency to a drive-line system that operates over a wide range of speed and power. In the vehicle simulation calculations made using HYVEC, component efficiency is not used, but instead the losses are calculated at each component operating point encountered in the driving cycle and added to the output needed to obtain the input power required for each component in the drive-line.

Table 3-16

ELECTRIC DRIVE SYSTEM EFFICIENCY AT HIGH LOADS

System Type	Efficiency (%) Motor	Controller	Combined
DC Separately Excited	88	97	85
AC Induction	93	94	87
AC PM Synchronous	92	93	85

In comparing dc and ac drive systems several conclusions seem to follow from Table 3-15 and Figures 3-8, 3-9. First, ac motors are more efficient than dc motors by a significant margin (4 to 6 percentage points). Secondly, the control of power to and from the motor in the vehicle application (battery electrical storage) is more efficient in the dc system. This is true both when the armature chopper is being used and even more so when it is bypassed above base speed. The net efficiency advantage of the ac drive system is probably less than two or three percentage points.

3.3.2.3 Motor and Controller Costs

Determination of OEM unit costs ($/kW) for the various electrical components is difficult for several reasons:

1. At least a rough preliminary design of the components is needed to estimate material and labor costs.

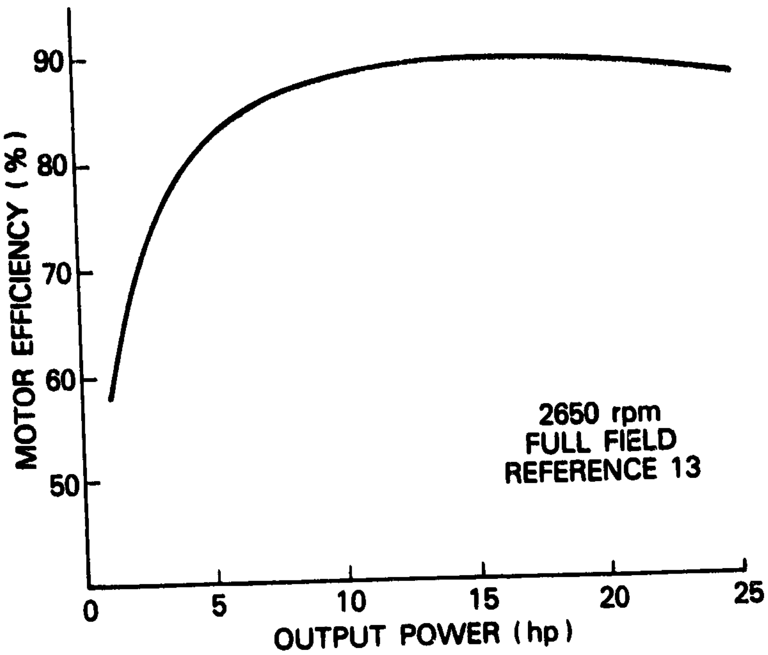

Figure 3-8. Efficiency of a dc Separately Excited Motor as a Function of Load

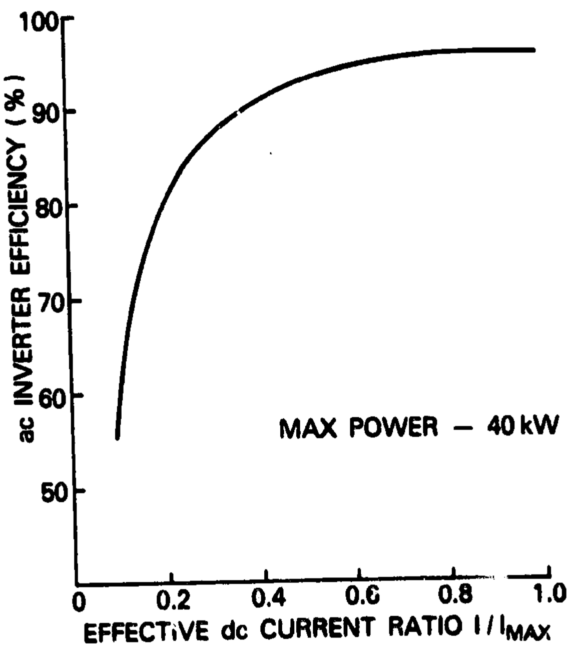

Figure 3-9. Efficiency of a Pulse Width Modulated ac Inverter as a Function of Effective dc Current

2. None of the components and few of the individual parts are presently sold in the high volume ($\geq 10^5$/yr) associated with the automobile industry.
3. One of the key components in the power conditioner, the power transistor, is currently only in an advanced state-of-development, and its cost is quite uncertain.

Some of these difficulties have been circumvented by using information/data that has been developed at General Electric Corporate Research and Development on other Department of Energy electric vehicle and advanced drive-line component contracts. In particular, a manufacturability study in connection with the Near-Term Electric Vehicle Program has recently been completed, and much of the information produced for the dc drive system is directly applicable to the hybrid vehicle program. In that study detailed cost data (materials and labor) were obtained for each component in the drive-line and the manufacturing cost of each component was estimated. Considerable simplification of the component design, including electronic circuits and microprocessor packaging, was done to reduce cost. Preliminary circuit design for the ac inverter was available to the hybrid program from the Advanced Controller Study (Contract No. DEN 3-59) for NASA-Lewis. It was then possible to use the same part costs for the ac inverter that were used in estimating the manufacturing cost of the dc armature controller. Direct current and alternating current motor costs were estimated based on information obtained from General Electric divisions which manufacture dc and ac motors.

The OEM price of the various electric drive components were calculated as follows:

$$OEM = \underbrace{1.2}_{\text{10\% profit after taxes}} \times \underbrace{1.1}_{\text{handling charge of vehicle fabricator}} MC$$

where MC $\equiv$ manufacturing cost

If the selling price of a complete component, such as a motor, is known, then

$$OEM = 1.1 \times SP$$

where SP = selling price to the vehicle fabricator

Based on the cited General Electric Corporate Research and Development studies, the following electrical component costs were determined for an electric drive system having a continuous rated power of 20 kW and a peak-rated power of 40 kW:

Direct Current System

- Manufacturing cost ($)

  Field chopper - 97

Armature chopper - 230 + 4 $(\$/M)_{TR}$
Microprocessor - 116
Battery charger - 191

- Selling price ($)

  dc separately excited motor - 540

- OEM price ($)

  Field chopper - 128
  Armature chopper - 304 + 5.3 $(\$/M)_{TR}$
  Microprocessor - 153
  Battery charger - 252
  dc separately excited motor - 595

- Specific costs ($/kW)

  Field chopper - 3.2
  Armature chopper - 7.5 + 0.133 $(\$/M)_{TR}$
  Microprocessor - 3.8
  Battery charger - 14 ($/kWh)
  dc motor - 29.75

Note that $(\$/M)_{TR}$ is the power transistor manufacturing cost per module - each module rated at 400 V and 250 A.

The corresponding information for the ac inverter and motor are the following:

Alternating Current System

- Manufacturing cost ($)

  ac inverter - 300 + 11.25 $(\$/M)_{TR}$

- Selling price ($)

  ac induction motor - 365

- OEM price ($)

  ac inverter - 396 + 14.9 $(\$/M)_{TR}$

- Specific costs ($)

  ac inverter - 10 + 0.37 $(\$/M)_{TR}$
  ac induction motor - 20

For the ac drive system, the costs of the microprocessor and the battery charger were taken to be the same as for the dc system.

The cost values just developed were taken to correspond to a production volume of 100,000 units/yr in 1978 dollars. It was assumed that if the production rate were increased to 1,000,000 units/yr, all the electrical component costs would decrease by 33%. In most comparisons between the selling price of the hybrid/electric and the reference ICE vehicles, the high-volume OEM prices were used. It was assumed that the heat engine and transmission costs discussed in Section 3.1 already corresponded to automotive industry production rates (i.e., 10^6 units/yr).

Calculations were made to show the effect of power transistor module cost on the incremental cost of the hybrid drive-line to the consumer (a dealer markup of 30% was assumed). The results are shown in Figure 3-10 for a hybrid vehicle using an electric drive-line with a continuous rating of 20 kW and a peak rating of 40 kW. Three electric drive-line options were considered:

1. Direct current motor, field control, battery switching
2. Direct current motor, armature control and field control
3. Alternating current induction motor with transistorized inverter

The calculations were made for power module costs between $25 and $100. The present best estimate is that a module cost of $50 is likely. The significant effect of module cost on the relative costs of the power train options is evident from Figure 3-10. It is clear that the cost penalty using the ac drive system becomes quite large if the power transistor cost is more than $50.

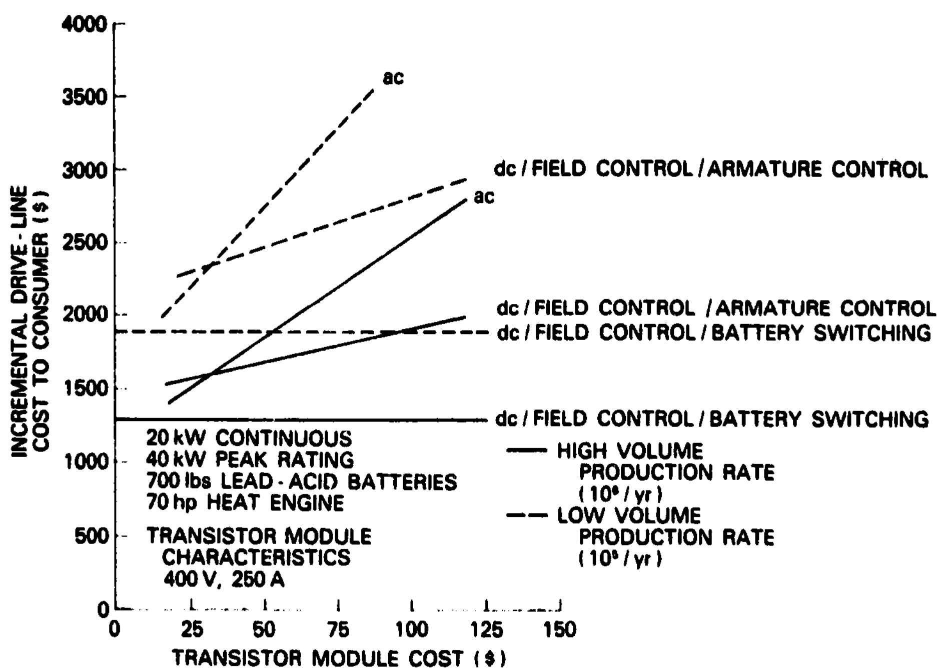

Figure 3-10. Effect of Power Transistor Module Cost on Incremental Drive-line Cost for Hybrid Vehicle

3.3.3 DETAILED ELECTRIC DRIVE SYSTEM CHARACTERISTICS

Detailed models of the electric drive systems (both ac and dc) have been developed for simulating the performance of the electric motor and controller subsystem on the various driving cycles. During the initial trade-off and preliminary design studies, the exact size of the motor drive system was not known. Therefore, the modelling approach used had to be flexible enough to allow the use of different size motors. Two reference motors were chosen, one dc separately excited and the other, an ac induction motor. The data on these motors are shown in Tables 3-17 and 3-18.

Table 3-17

REFERENCE DC MOTOR DATA

Continuous Duty Rating	-	20 hp
Base Speed	-	2500 rpm
Maximum Speed	-	5000 rpm
Rated Current	-	175 A
Rated Voltage	-	96 V
Rated Field Current	-	4.9 A
<u>Armature Winding</u>		
Winding Resistance	-	0.024 Ωm
Winding Inductance (unsaturated)	-	0.52 mH
Turns per Pole	-	5.88
<u>Field Winding</u> - Separately Excited		
Winding Resistance	-	4.3 Ωm
Winding Inductance (unsaturated)	-	2.3 H
Turns per Pole	-	330
Torque Constant K_t	-	0.352 lb-ft/A-megaline
Voltage Constant K_V	-	0.05 V/rpm megaline

The reference motor parameters have been normalized so that they can be used for any motor whose horsepower is in the range of 10 to 30 hp continuous duty. The normalization of the motor parameters is carried out using the rated voltage, current, speed, and flux of the reference motor. In order to use the motor model, the rated output power, speed, flux, and voltage of the electric motor in the vehicle must be specified.

To illustrate the scaling methodology used, consider the dc motor model equations given by:

<u>Armature Voltage</u>

$$V_a = K_V \omega \phi + V_{bd} + (R_a + R_L)\ I_a \tag{1}$$

<u>Motor Developed Torque</u>

$$T_e = K_t \phi I_a \tag{2}$$

Table 3-18

REFERENCE AC MOTOR DATA

Continuous Duty Rating	-	20 hp
Base Speed (60 Hz)	-	1747 rpm
Voltage per Phase (LN)	-	266 V
Line Current	-	23.32 A
Power Factor	-	0.87
Ship	-	0.0297
Stator Resistance (per Phase at 95 °C)	-	0.3322 Ωm
Rotor Resistance (per Phase Referred to Stator)	-	0.2466 Ωm
Stator Reactance	-	1.157 Ωm
Rotor Reactance (per Phase Referred to Stator)	-	2.184 Ωm
Magnetizing Reactance (per Phase Referred to Stator)	-	42.45 Ωm
Magnetizing Branch Resistance (in Series with Reactance)	-	1.467 Ωm

Motor Losses

- Friction losses

$$P_{FL} = C_{FL}\ \omega \tag{3}$$

- Stray load

$$P_{SL} = 0.01\ P_M \tag{4}$$

- Core losses

$$P_{CL} = C_{CL}\ \omega^{1.5} \tag{5}$$

- Windage losses

$$P_{WL} = C_{WL}\ \omega^{3} \tag{6}$$

Using the base quantities V_B, ϕ_B, I_B and T_B, equations 1 through 6 are transformed into unitized form as follows:

$$\frac{V_a}{V_B} = \frac{K_V \omega_B \phi_B}{V_B} \cdot \left(\frac{\omega}{\omega_B}\right)\left(\frac{\phi}{\phi_B}\right) + \frac{V_{bd}}{V_B} + \left(\frac{R_a + R_L}{V_B}\right) \cdot I_B \left(\frac{I_a}{I_B}\right) \tag{7}$$

$$\frac{T_e}{T_B} = \frac{K_t \phi_B \omega_B}{V_B * f_c} \tag{8}$$

$$\frac{P_{FL}}{V_B I_B} = \frac{C_{FL}\omega_B}{V_B I_B}\left(\frac{\omega}{\omega_B}\right) \qquad (9)$$

$$\frac{P_{FL}}{V_B I_B} = 0.01\ \frac{P_M}{V_B I_B} \qquad (10)$$

$$\frac{P_{CL}}{V_B I_B} = C_{CL}\ \frac{\omega_B^{1.5}}{V_B I_B}\left(\frac{\omega}{\omega_B}\right)^{1.5} \qquad (11)$$

$$\frac{P_{WL}}{V_B I_B} = C_{WL}\ \frac{\omega_B^{3}}{V_B I_B}\left(\frac{\omega}{\omega_B}\right)^{3} \qquad (12)$$

where

$$T_B = \frac{V_B I_B}{\omega_B}\cdot f_c \qquad (13)$$

$$f_c = \text{Conversion Factor} = 7.047\ \frac{\text{ft-lb rpm}}{\text{watts}} \qquad (14)$$

The motor parameters defined by the following equations are assumed to be constant for all motors in the horsepower range used in the trade-off and preliminary design studies.

$$K_V' = K_V\ \frac{\omega_B \phi_B}{V_B} \qquad (15)$$

$$V_b'{}_d = \frac{V_{bd}}{V_B} \qquad (16)$$

$$R_a' = \frac{R_a I_B}{V_B} \qquad (17)$$

$$R_L' = \frac{R_L}{V_B}\cdot I_B \qquad (18)$$

$$K_t' = \frac{K_t \phi_B \omega_B}{V_B \cdot f_c} \qquad (19)$$

$$C_{f\ell}' = \frac{C_{F\ell}\omega_B}{V_B I_B} \qquad (20)$$

$$C_{c\ell}' = C_{C\ell}\cdot \frac{\omega_B^{1.5}}{V_B I_B} \qquad (21)$$

$$C_{WL}' = C_{\omega L} \frac{\omega_B^{\;3}}{V_B I_B} \qquad (22)$$

For example, to obtain the actual value of the vehicle motor parameters $K_{V_{veh}}$, using equation 14 gives

$$K_{V_{veh}} = K_{V_{ref}} \frac{\omega_{B_{ref}} \phi_{B_{ref}}}{V_{B_{ref}}} \cdot \frac{V_{B_{veh}}}{\omega_{B_{veh}} \phi_{B_{veh}}} \qquad (23)$$

The dc motor model has two controller options -- separate armature and field chopper circuits or a field chopper circuit with a starting resistor in series with the armature windings. There is also an option for battery switching with the field chopper circuit. In addition, models developed are used for regenerative braking.

The dc motor model equations are the same as those used for the simulation of the GE/DOE electric car. The difference is in the computational approach. The algorithm for the GE/DOE electric car simulation program starts with an estimate of the armature current and iteratively improves upon the estimate until convergence is reached using a prespecified tolerance. In the hybrid vehicle simulation program, however, it is necessary to reduce the computation time because a large number of simulations for long driving cycle distances (time) had to be performed for the various candidate configurations and drive-line component combinations. Therefore, iterative techniques requiring long computational time have been avoided as much as possible.

The nonlinear battery model equations are linearized and combined with the motor equations and solved directly to obtain the battery terminal voltage and current. Since the battery model is updated at every iteration, the effect of the linearization on accuracy is small.

A plotting subroutine has been added to the simulation program which allows the plotting of 20 selected variables. Between one and five variables can be plotted in one frame. Since the simulation for any position of the driving cycle can be plotted, detailed study of any portion of the driving cycle is possible.

Figures 3-11 to 3-16 show some of the hybrid vehicle variables for the EPA urban driving cycle. Table 3-19 gives the data on the hybrid vehicle propulsion system. For this run, the heat engine is the primary drive for vehicle speed above 31 mph and the electric motor is the primary drive for speeds below 31 mph. Figure 3-12 shows instances when both heat engine and motor must be on because the primary drive alone cannot supply all the power (as in points A, B and C). At point C on Figure 3-12, the vehicle was traveling at 36 mph with an acceleration of 2.8 mph/s. The total power required to drive the vehicle is 72 hp. Since the maximum engine power was only 47 hp, the motor had to be turned on. With both propulsion systems operating the power required was split equally between them.

A listing of the vehicle simulation program (HYVEC) is given in Volume III. The program using the dc electric drive system has been run routinely. For an urban driving schedule (10 EPA urban cycles), the computing time is five to ten minutes on the Honeywell 605 in the time-sharing mode. The motor subroutine for an ac induction motor has been prepared, but that option of the program has not been checked out.

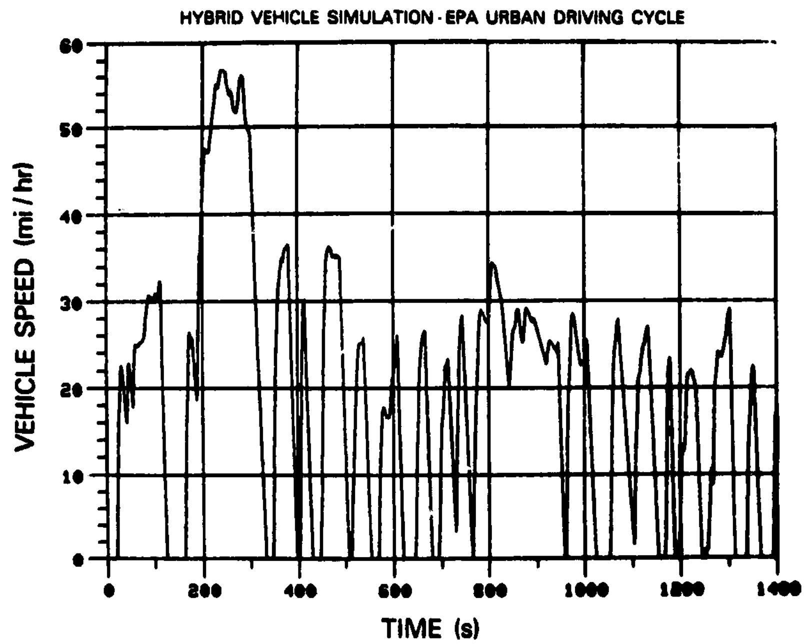

Figure 3-11. Vehicle Speed, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

Table 3-19

HYBRID VEHICLE SIMULATION DATA

(Total Vehicle Weight (+batteries) = 4,000 lbs)

Engine

Peak Power	-	73 hp
Type	-	VW 1.6-ℓ, gasoline
Idle Speed	-	900 rpm
Maximum Speed	-	6,000 rpm

DC Motor - Field control only with battery switching

Rated Output	-	24.4 hp
Peak Output (Short Time)	-	48 hp
Base Speed	-	2,000 rpm
Maximum Speed	-	6,000 rpm
Rated Voltage	-	96 V
Rated Current	-	213 A
Rated Flux	-	0.9 megaline
Rated Field Current	-	5.88 A

Transmission - Four-speed automatically shifted

Gear Ratios	-	3.46, 1.94, 1.29, 1.0
Axle Ratio	-	3.3

Battery Lead-Acid

Ampere-hour, Capacity		110 at C/3 rate
Number of Cells per module	-	3
Total Number of Modules	-	18

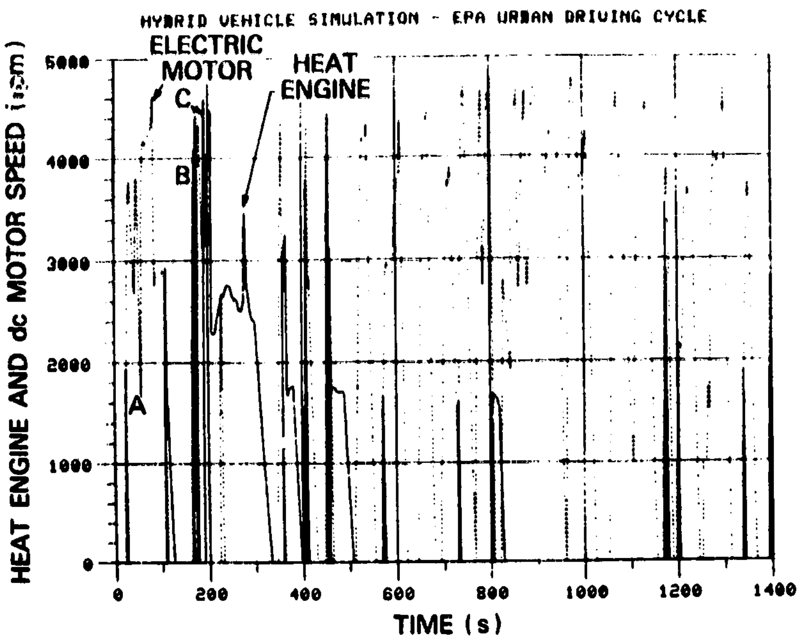

Figure 3-12. Heat Engine and dc Motor Speed, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

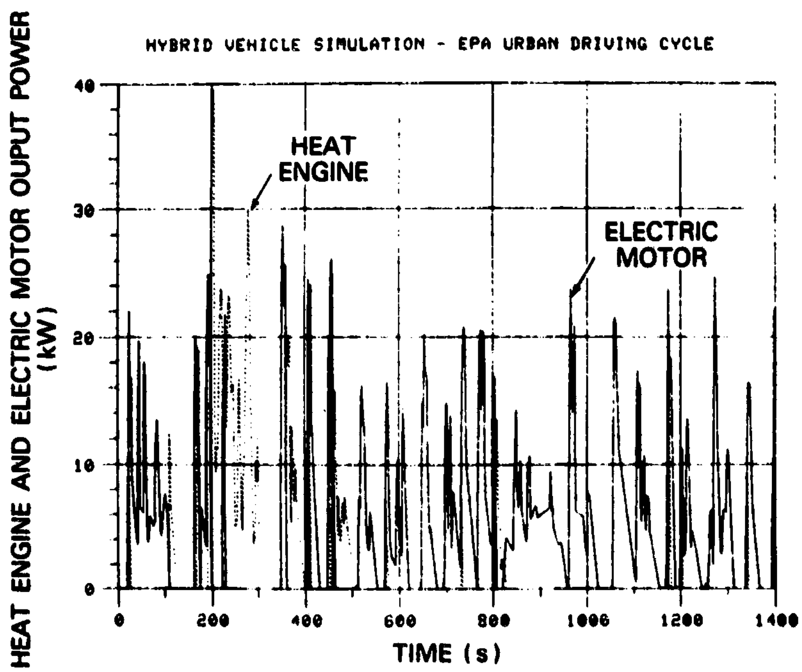

Figure 3-13. Heat Engine and Motor Output Power, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

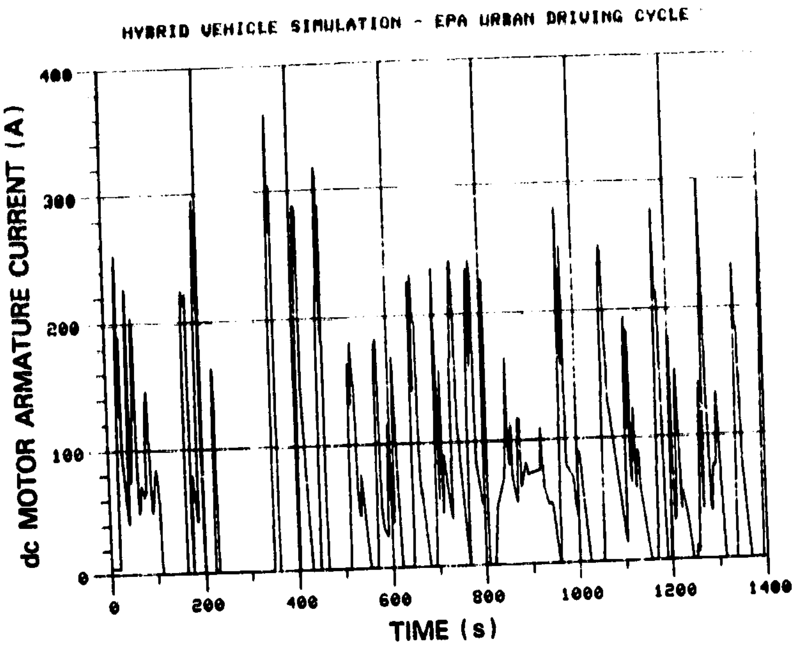

Figure 3-14. DC Motor Armature Current, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

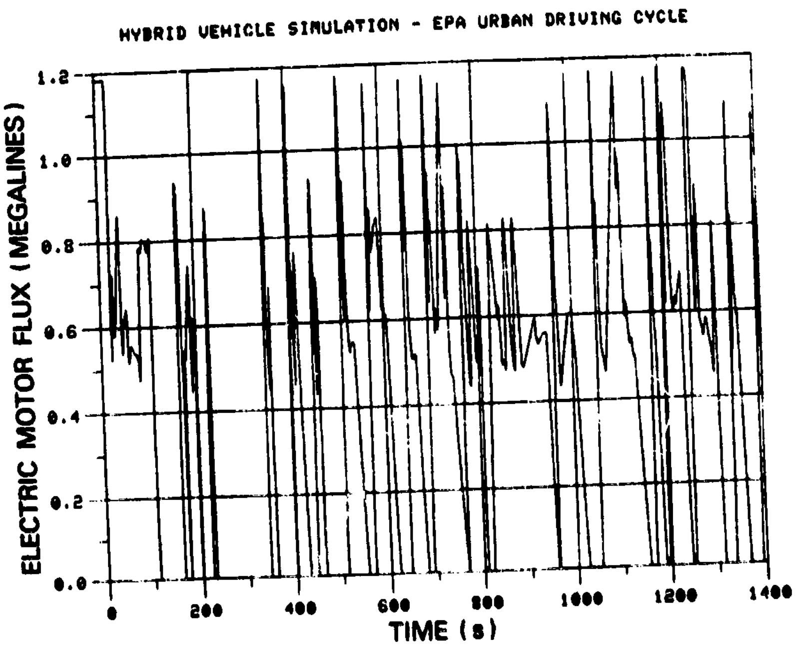

Figure 3-15. Electric Motor Flux, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

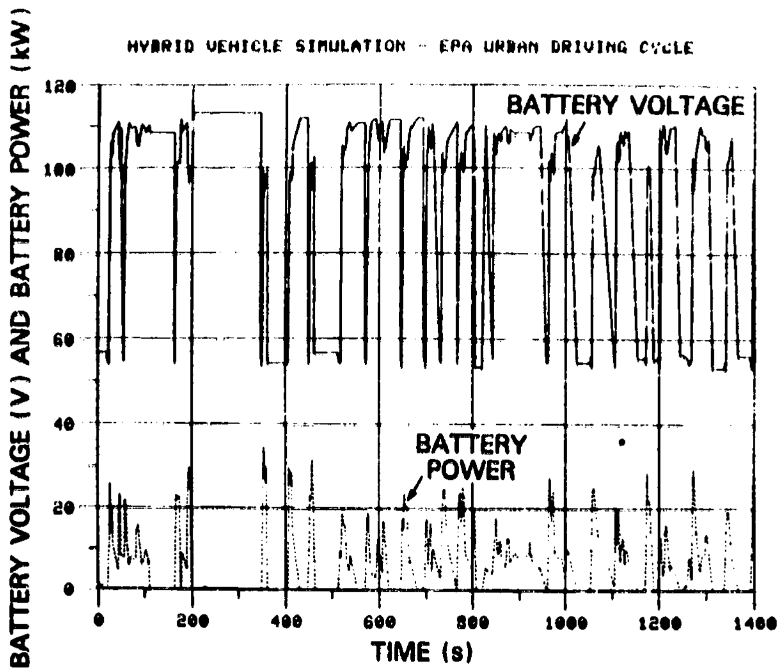

Figure 3-16. Battery Voltage and Battery Power, Hybrid Vehicle Simulation, EPA Urban Driving Cycle

3.4 BATTERIES FOR HYBRID/ELECTRIC VEHICLES

3.4.1 TYPES OF BATTERIES CONSIDERED

Work is currently being done on a relatively large number of secondary battery (rechargeable) systems for storing electricity for both vehicle and stationary applications. A summary of some of the most promising of those battery development activities for vehicle application is given in Table 3-20. A number of organizations are working on each battery system. As part of the present study, ESB Technology performed a survey of the present and projected status of the various battery systems. The results of that survey are included in the battery supplement to this report contained in Volume II. In addition, information on the status of battery technology in Japan was prepared by Diahatsu Motor Company under a separate consulting agreement with General Electric. That information is also included in the battery supplement.

In the present study, major attention was focused on the lead-acid, Ni-Zn, and Ni-Fe battery systems because the ESB survey indicated that the high-temperature Li-S and Na-S batteries would not be available for Near-term vehicle use in the 1981 to 1985 time period. A high-temperature battery category is included in Section 3.4.3 and in the discussion of hybrid vehicle synthesis (Section 5). High-temperature batteries are not included in the second-by-second vehicle simulations discussed in Section 8.

3.4.2 HYBRID VEHICLE BATTERY REQUIREMENTS

Before considering the specific characteristics (e.g., Wh/lb, W/lb, cycle life, $/kWh, etc.) of the various types of batteries, battery requirements for the hybrid/electric application are discussed and, in particular, important differences between the use of the batteries in the all-electric and hybrid/electric applications are highlighted. For the all-electric vehicle, the battery pack is larger (between 25 to 30% of the vehicle weight) than for the hybrid/electric and it must provide all the energy to drive the vehicle. In the case of the hybrid/electric vehicle, the battery pack is smaller (between 15 to 20% of the vehicle weight) and is supplemented by the heat engine (i.e., when the battery cannot provide the required power, the heat engine is available to pick up the load). These differences between the all-electric and hybrid/electric applications have the following implications in terms of battery use and requirements.

1. The average discharge rate of the batteries in a hybrid is 30 to 50% greater than in an all-electric vehicle (i.e., the batteries in the hybrid are likely to be discharged at the C/1.5-C/2 rate rather than the C/2.5-C/3 rate as in an all-electric vehicle.

Table 3-20

CANDIDATE BATTERIES FOR HYBRID/ELECTRIC VEHICLES

Type	Active Companies	Status	Advantages	Disadvantages
Lead-Acid	Globe-Union Eagle-Picher ESB	Commercially Available, Vehicle Testing and Laboratory Testing of Improved Cells/Batteries	High Power Density, Relatively Easy to Charge and Maintain, Technology Exists and Is Being Improved, Reasonable Cost	Relatively Low Energy Density
Ni-Zn	Energy Research Corporation Gould Yardley GM	Limited Vehicle Testing, Primarily Laboratory Testing to Improve Life	Relatively High Energy and Power Density	Limited Life and High Cost
Ni-Fe	Westinghouse Eagle-Picher Matsushita, Japan	Limited Vehicle Testing in Japan, Primarily Laboratory Testing of Cells in the US	Relatively High Energy Density, Long Life	Relatively Low Power Density, Difficult to Charge and Maintain, High Cost to Date
LiAl/FeS_x	Eagle-Picher Gould, Both Under Argonne NL	Laboratory Testing of Cells, Initial Work on Cell Modules	Potential for High Energy Density, Projected Low Cost	High Operating Temperature, Relatively Low Power Density, Overcharging Decreases Cycle Life
Na-S	Ford Motor Company Chloride (England) British Railways (England)	Laboratory Testing of Cells, Initial Work on Cell Modules and Batteries	Potential for High Energy and Power Density	High Operating Temperature, High Cost of Beta Alumina Electrolyte, Not Tolerant of Freeze-Thaw Cycles, Difficulty in Sending to Large AH Cells

2. The peak power required per unit weight of battery is higher in a hybrid vehicle than in an all-electric vehicle both because the battery pack is a smaller fraction of the vehicle weight and the power-to-weight ratio of the hybrid is much higher (0.02 kW/lb compared with 0.009 kW/lb for the electric vehicle).
3. The batteries in the hybrid will be deep discharged a greater fraction of the days because the electric range of the hybrid vehicle will be about one-half that of the all-electric car.
4. The increased discharge rate means that the average heat dissipation in the battery pack will be considerably greater for the hybrid which will result in the batteries operating at higher temperature unless they are cooled.
5. If the hybrid and all-electric drive systems operate at the same voltage, the size of the cells (ampere-hours) needed for the hybrid application is 30 to 50% smaller than the size of those needed for the all-electric car.

All the above implications (one through five) mean that the batteries in the hybrid/electric vehicle have a potentially more severe duty cycle than those in the all-electric car. The only saving feature of the hybrid application is that the heat engine can be used to assist and load-level the electric drive (i.e., the batteries) when desired. However, as the heat engine carries a greater fraction of the load, more of the energy to power the car comes from gasoline and less electrical energy is substituted for petroleum fuel. Since the objective of the hybrid/electric design is to use primarily electrical energy for urban travel at least up to the electric range of the hybrid vehicle, it appears inevitable that the duty cycle for batteries in a hybrid will be more severe than in an all-electric. It also seems inevitable that as the batteries age and are incapable of providing full energy and power, the heat engine will be forced to carry a greater fraction of the load, and the fraction of electricity substituted for gasoline will decrease.

3.4.3 SPECIFIC CHARACTERISTICS

3.4.3.1 Energy and Power Density

As far as vehicle performance (range and acceleration) are concerned, the key battery characteristics are energy density (Wh/lb) and power density (W/lb). As discussed in Reference 5, the battery pack for an electric or hybrid-electric vehicle can be sized by either range or power requirements depending on the design range and power-to-weight ratio of the vehicle. Unfortunately, neither energy density nor power density are fixed values for a given battery and depend on the discharge rate (average current and peak current, respectively). In addition, the maximum usable power density depends on the voltage droop that can be tolerated by the electric drive system. The power density corresponding to one-half open-circuit cell voltage for the battery is of little use as the resultant battery pack voltage is so low that the electric motor cannot provide the power required. In general, a voltage droop of more than 20 to 25% is not useful in vehicle applications. Another point-of-interest in discussing maximum battery power density is the time period for which the battery must sustain the high power. Since most traction motors can sustain peak power (about twice the rated continuous power) for only about one minute, a battery need only supply its peak power for one minute also. Maximum power density (W/lb) of a battery is often stated separately for steady-state and short-pulse operation. Steady-state operation refers to vehicle gradability or top speed and short pulse to the maximum effort of 0 to 30 mph or 0 to 45 mph accelerations. Based on motor thermal limitations, gradability/top speed can be sustained for only 60-s, and vehicle performance specifications require acceleration times of 10 to 20 s. Hence, a short-pulse high-power for the battery is about 15 s and a long pulse high-power is about 60 s. Both power pulses must be possible with a voltage droop of less than 20 to 25%.

Energy and power density characteristics of various types of vehicle traction batteries are given in Table 3-21. Energy density is given for the three-hour rate, and power density is given for both 15-s and 60-s power pulses - both for a battery which is 50% discharged. The values given in Table 3-21 are based on the battery data in the battery supplement in Volume II. It has been assumed that the power density for a 60-s pulse can be determined directly from the voltage-current discharge data. The power density for the short 15-s pulse was based on the 60-s power density and engineering judgment. Pulsed discharge tests of lead-acid batteries at various states of discharge are currently under way at ESB. Data from those tests is given in Volume III. Characteristics are shown in Table 3-21 for both Improved State-of-the Art (ISOA) and advanced-projected batteries. ISOA designates batteries which would result from the further development and refinement of existing design/fabrication approaches and does not require any technology breakthroughs. Except for the Li-S batteries, ISOA batteries could be available in the time period of the Near-Term Hybrid Vehicle (NTHV) Program. "Advanced-projected" means that to attain the battery characteristics indicated new design/fabrication approaches are needed, and, as a result, there is considerable uncertainty when and if those characteristics can be achieved. Advanced-projected batteries almost certainly will not be available for the NTHV program.

As noted previously, the average discharge rate for batteries used in a hybrid vehicle will be 30 to 50% greater than in an all-electric car. Hence, it is of interest to consider the effect of discharge rate on the ampere-hour capacity or energy density of the various types of batteries. This is shown in Figure 3-17. The effect of discharge rate is particularly significant for the lead-acid and Li-S batteries. It should also be noted that since the cell ampere-hour capacity of the hybrid vehicle batteries will be smaller than for the corresponding all-electric vehicle batteries, it will be more difficult to attain as high an energy density in

Table 3-21

ENERGY AND POWER DENSITY CHARACTERISTICS
OF ELECTRIC/HYBRID VEHICLE BATTERIES

	$(Wh/lb)_{C/3}$	$(W/lb)^*_{15s}$	$(W/lb)^*_{60\ s}$
Lead-Acid (ISOA)	18	65	55
Lead-Acid (Adv-projected)	27	85	75
Ni-Zn (ISOA)	30	85	75
Ni-Fe (ISOA)	30	50	40
Li-S (ISOA)	40	40	35
Li-S (Adv-projected)	50	80	65

*Battery 50% discharged

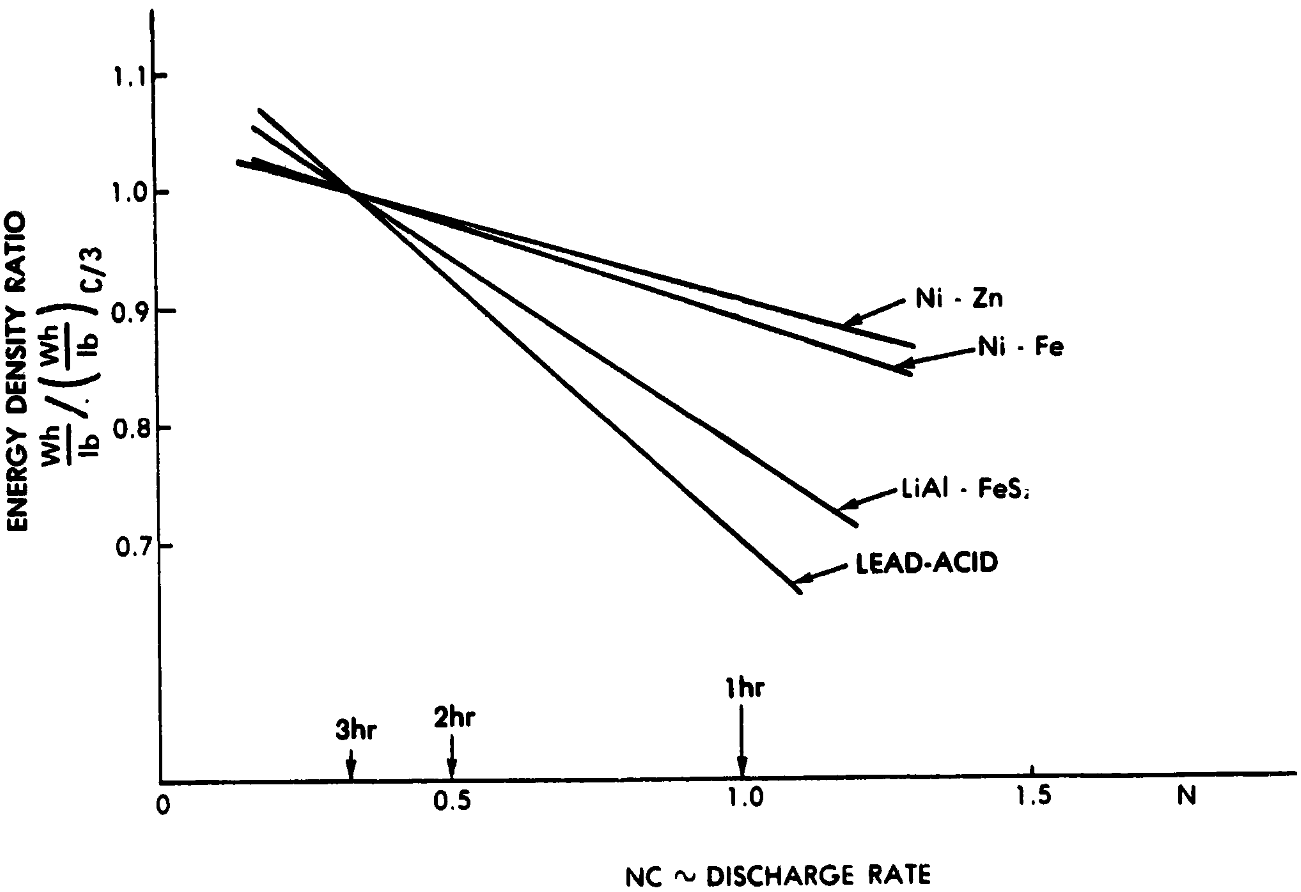

Figure 3-17. Effect of Discharge Rate on the Energy Density of ISOA Batteries

the hybrid vehicle batteries as is currently attained in batteries designed for electric vehicles. This penalty in energy density is at least partially off-set by the inherent improvement in power density which results from the need for smaller ampere-hour capacity in the hybrid vehicle batteries. These differences between hybrid and all-electric battery requirements undoubtedly mean that it will be necessary to design/fabricate new cells/battery modules for the hybrid vehicle application.

3.4.3.2 Battery Costs and Lifetime

Available information on battery costs and lifetime (cycle life for deep discharges) is very "soft" and uncertain. The only batteries commercially available for vehicle use are lead-acid batteries for golf-carts, and they are produced in relatively large quantities (> 100,000/yr). The current replacement price for an ESB EV-106 battery (bought in quantities of 250) is $45 or about $50/kWh. Current projections of battery cost ($/kWh or $/lb) are necessarily uncertain because of fluctuations in material costs (e.g., Pb, Ni, separators, etc.), fabrication techniques, and degree of automation in the production facilities. As far as cost is concerned, all that can be done at the present time is to list the claims/goals of the various battery groups. This is done in Table 3-22. When available, the present cost of the battery type is also noted in the table. The cost values given represent OEM costs to the hybrid vehicle manufacturer. Hence, to those costs must be added the vehicle manufacturer handling cost and dealer markup.

In determining the life cycle operating cost of the battery to the hybrid vehicle owner, battery lifetime is equally as important as initial or replacement cost. Battery lifetime is usually given in terms of number of charge/discharge cycles to a prescribed depth of discharge. This is a measure of the total AH (or kWh) that can be used from the battery over its lifetime. In actual use, vehicle traction batteries are not deeply discharged between each charge. Therefore, the total number of charge/discharge cycles in practice would have to be greater than the stated lifetime in deep discharges if the battery is to achieve the stated life cycles operating cost. Batteries used in hybrid vehicles would undoubtedly experience more regular deep discharges than those used in all-electric cars. Cycle life data for commercially available ESB EV-106, State-of-the-Art (e.g., Globe-Union EV2-13) or Improved State-of-the-Art batteries are very limited. In addition, the relevancy of laboratory battery cycle life data to cycle life of battery packs in actual field operation is open to question (Reference 16). Nevertheless, within the above stated limitations and uncertainties the estimated attainable cycle life for the various battery types is given in Table 3-22. In nearly all cases, the cycle life given represents the goals of the various battery research and development programs and have not as yet been demonstrated by either laboratory or field tests on large numbers of units.

Table 3-22

COST AND CYCLE LIFETIME CHARACTERISTICS
OF VARIOUS TYPES OF BATTERIES

Battery Type	Cost ($/kWh)* Present	Cost ($/kWh)* Goal	Cycle Life† Present	Cycle Life† Goal
Lead-Acid	40-50	50	300	1000
Ni-Zn	>150	60	<200	>800
Ni-Fe	>100	60	1500	2000
Li-S (FeS_2)	—	40	<300	>800

*OEM cost
†For 80% depth of discharge

Irrespective of the uncertainties regarding battery cost and lifetime, a number of statements can be made with good confidence:

1. ISOA lead-acid batteries costing about $50/kWh and having a cycle life of 500 to 800 cycles are expected to be available within several years.
2. Ni-Zn batteries are currently quite expensive (> $150/kWh) and have limited cycle life (< 200 cycles). Dramatic reductions in cost and improvements in cycle life are required before Ni-Zn batteries can be considered for use in mass-marketed hybrid/electric cars.
3. Ni-Fe batteries have long cycle life (> 1000 cycles) and relatively high cost (> $100/kWh). Based on a combination of cost and cycle life, Ni-Fe batteries may have lower lifetime operating costs than lead-acid batteries. A dramatic reduction in cost is needed before the Ni-Fe battery will be clearly more attractive than lead-acid for hybrid/electric vehicles.

3.4.3.3 Charging and Maintenance

For vehicle applications, the charging and maintenance requirements of the battery systems are particularly important because, placed in the hands of the general public, battery charging must be simple and safe and battery maintenance must be relatively infrequent and convenient to perform. As indicated in Table 3-23, the charging and maintenance characteristics of the various types of batteries are quite different and in some cases present considerable difficulty for use in passenger cars.

The characteristics of lead-acid batteries seem to be compatible with their use by the general public. Charging is relatively simple, safe, and efficient especially with a tapered-current charger. Venting and maintenance (primarily battery watering) are acceptable using a single-point, automatic watering system. Periodic battery equalization charging may or may not be advantageous from a battery life standpoint, but it apparently is not essential.

Charging of Ni-Zn batteries is not as simple as that of lead-acid. The Ni-electrode requires overcharge to get full capacity from the battery, and overcharge of the Zn-electrode is both injurious to the zinc electrode and hazardous due to hydrogen formation. To circumvent these problems requires careful design of the zinc electrode and special charging procedures including a reliable indicator of the state-of-charge of the Ni-electrode. This approach has been taken by Energy Research Corporation (Reference 17) and has resulted in satisfactory charging of their Ni-Zn battery pack. Careful control of the overcharge probably results in watering requirements for the Ni-Zn battery which are not much different from those for lead-acid. Another approach to charging Ni-Zn batteries is to design a sealed cell in which the oxygen generated at the Ni-electrode during the charging is recombined with the Zn-electrode to maintain the Zn-electrode at less than

Table 3-23

CHARGING AND MAINTENANCE CHARACTERISTICS OF VARIOUS TYPES OF BATTERIES

Battery Type	Charging		Maintenance	
	Efficiency %	Problems	Problems	Effect of Ambient Temperature
Lead-Acid	75	None	Relatively Infrequent Watering	Moderate
Ni-Zn	75	Overcharge Required, Difficult to Determine State-of-Charge	Relatively Frequent Watering	Moderate
Ni-Fe	60	Cooling Required	Frequent Watering	Large
Li-S	75	Overcharging Shortens Life; Cell Equalization Required	System Operates at 400 °C	None (System Thermally Insulated)

a full state-of-charge. This concept is utilized in sealed Ni-Cd cells and is the approach being taken by General Motors (GM) (References 18,19) to develop sealed Ni-Zn/batteries for vehicle use. Clearly the charging and maintenance characteristics of the sealed cells would be very attractive if and when they are developed. Relatively little information is available on the status of the GM Ni-Zn program and how close they might be to the development of a sealed Ni-Zn battery with satisfactory lifetime (> 500 cycles).

As noted in the battery supplement in Volume II, Diahatsu Motor Company in Japan has tested an electric subcompact car using Ni-Fe batteries. The batteries functioned satisfactorily in the vehicle (i.e., provided the desired range and power) but were found to be unsatisfactory as far as charging and maintenance were concerned. The Ni-Fe batteries had to be charged carefully and cooled to prevent overheating. They had to be watered about once a week. Gassing takes place throughout the charging process for Ni-Fe cells and the batteries must be carefully vented. Ni-Fe batteries also have poor low-temperature performance. For example, an electrolyte temperature of 0 °C yields only 43% of its 25 °C capacity.

The handling of LiAl-FeSx batteries is much different from that of the other systems because the Li-S cells operate at high temperature - 400 to 450 °C and use nonaqueous electrolytes. The batteries are placed inside a thermally controlled jacket which reduces the average heat loss to about 200 W and maintains the battery near its operating temperature during periods of prolonged vehicle inactivity. When it does become necessary to cool the battery (i.e., freeze the molten electrolyte), the resultant freeze-thaw cycle does not seem to be injurious to the battery. Charging of a Li-S battery pack is complicated by the necessity to avoid overcharging any of the cells in the battery module. Overcharging produces sulfides which are injurious to the cell containers and can lead to cell failure. After the battery thermal control and charge equalization systems are developed, charging and maintenance of the LiAl-FeSx battery system should be simple, safe, and automatic as far as the vehicle owner is concerned.

3.4.3.4 Conclusions

It seems appropriate at this point to summarize the discussions of the previous sections and to rank the battery types qualitatively relative to their availability and acceptability for use in the Near-Term Hybrid Vehicle (NTHV) Program in 1982. This is done in Table 3-24. Some subjective judgments were clearly required in preparing Table 3-24, but when possible, such judgments were based on the vehicle design synthesis calculations discussed later in Section

As far as availability is concerned, it is expected that at least three types of batteries will be available for testing in the NTHV. Unfortunately, only the ISOA lead-acid battery comes

Table 3-24

SUMMARY OF THE PRESENT AND PROJECTED STATUS OF THE VARIOUS TYPES OF BATTERIES FOR HYBRID VEHICLE APPLICATIONS

Battery Type	Availability in 1982	Wh/lb	W/lb	Cycle Life	$/kWh	Charging and Maintenance
Lead-Acid						
ISOA	o	o(-)	o	o	o	o
Advanced	x	o	o	o	o	o
Ni-Zn (ISOA)	o	o	o	x	x	o
Ni-Fe (ISOA)	o	o	o(-)	o	x	x
Li-S						
ISOA	o(-)	o	x	o(-)	x	o
Advanced	x	o	o	o	o	o

Code: o - Acceptable
o(-) - Marginally acceptable
x - Not available or unacceptable

close to meeting all the performance, economic, and battery handling requirements for a mass-marketed hybrid-electric vehicle. As noted, the energy density (Wh/lb) of the ISOA lead-acid battery seems only marginally acceptable. Ni-Zn batteries have acceptable, in fact, attractive, energy and power density (W/lb) for the hybrid application, but considering their present state of development, it seems unlikely that their cycle life and cost will be acceptable for introduction in a mass-marketed vehicle by 1985. However, because of the attractiveness of their noneconomic characteristics, Ni-Zn batteries will be included in the second-by-second simulation studies discussed in Section 8.

Of all the batteries currently being tested in electric vehicles, the Ni-Fe battery appears to have the greatest cycle life. Depending on its initial cost, Ni-Fe batteries could also have the lowest operating cost (i.e., ¢/mi for battery replacement). Even though the maintenance and charging of Ni-Fe batteries present some difficulties, and the power density (W/lb) is low for the hybrid/electric application, the Ni-Fe battery will be included in the more detailed hybrid vehicle simulations.

The LiAl-FeSx batteries are not included in the detailed simulations because the projected power density of the ISOA designs are too low (probably less than 30 W/lb), and they offer no advantage in any respect over the more developed lead-acid, Ni-Zn, and Ni-Fe batteries. As discussed in a subsequent section, it seems likely that Li-S batteries would require the use of secondary energy storage, such as a flywheel, for effective use in electric or hybrid/electric vehicles having acceleration performance comparable to a conventional vehicle.

3.4.4 DETAILED BATTERY CHARACTERISTICS

In order to simulate the hybrid vehicle second-by-second on the various driving cycles, it is necessary to characterize the battery response in terms of terminal voltage as a function of battery current and state-of-charge. The characteristics of each type of battery are based on experimental data taken on a particular battery (Ah/cell) of that type, but must also be applicable to other batteries of that type having different cell capacities (Ah). Hence, in this section charge/discharge characteristics of a reference battery for each type are given, and the methodology is shown of how the reference parameters are used to determine the operating characteristics of batteries of differing Ah capacities.

3.4.4.1 Lead-Acid Batteries

The discharge characteristics used for the ISOA lead-acid battery are based on data from tests of the Globe-Union EV2-13, which is the battery developed for the Department of Energy-General Electric Near-Term Electric Vehicle. The voltage-current-state-of-charge characteristics of that battery are given in

Figure 3-18. State-of-charge (S) is defined in terms of the fraction of ampere-hours used for each average discharge current or

$$S = (Ah)\ \text{used}/(Ah)\ \text{capacity at } I_{avg}$$

The cell size for the reference battery is given for discharge at the C/3 rate. The cell rating for the Globe-Union battery is 170 Ah. The decrease in cell rating at higher discharge rates is given in Figure 3-19.

Figures 3-18 and 3-19 are used in the HYVEC program in the following manner to determine the operating characteristics of the vehicle battery which would have a different cell rating - say $(Ah)_v$ - than the reference cell. In the vehicle simulation program, the power and voltage required from the battery are determined by the electric motor subroutine. Hence, the battery subroutine is entered with knowledge of those quantities, and the objective is to determine whether the battery can supply that voltage at the required power level and, in addition, what is the resultant battery current. As indicated in Figure 3-19, the voltage-current characteristic of the lead-acid battery can be written as

$$E_B = E_{Bo} - R_B\ I_B \tag{1}$$

Where E_{Bo}, R_B are fractions of state-of-charge (S). Multiplying Equation (1) by E_B, one obtains

$$\left(\frac{E_B}{E_o}\right)^2 = \frac{E_{Bo}}{E_o}\left(\frac{E_B}{E_o}\right) - \frac{R_B}{E_o^2}\ P_B \tag{2}$$

where

$P_B \equiv$ battery power $= E_B\ I_B$

$E_o \equiv$ open circuit voltage of the reference battery

E_{Bo} and R_B in Equation (2) are known for the reference battery but not for the vehicle battery. It is assumed that Equation (2) applies to the vehicle battery at the same power density per cell. It is further assumed that $(Ah)_{capacity}$/cell is an accurate measure of cell weight. Hence, the power $(P_B)_{Ref}$ in the reference battery equivalent to the power required $(P_B)_v$ from the vehicle battery is given by

$$(P_B)_{Ref} = \frac{(P_B)_v}{(N_m)_{Ref}\ N_p N_s}\ \frac{(Ah)_{Ref}}{(Ah)_v} \tag{3}$$

where

$(N_m)_{Ref}$ = number of cells in the reference battery module

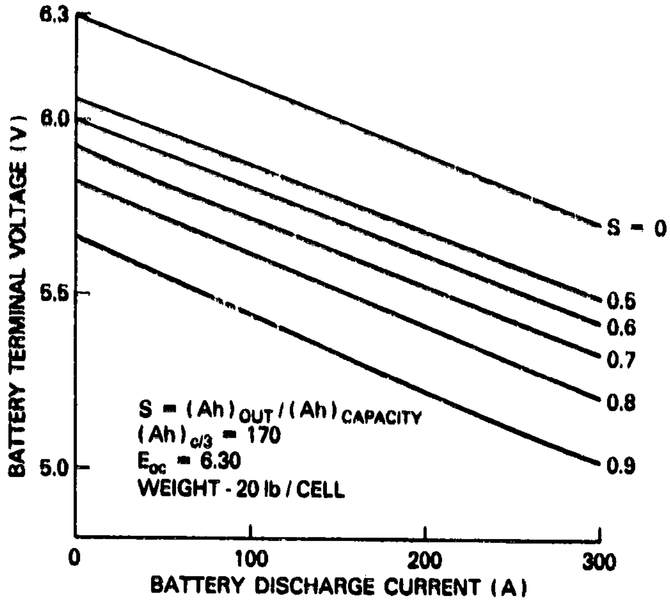

Figure 3-18. Lead-Acid Battery Discharge Characteristics

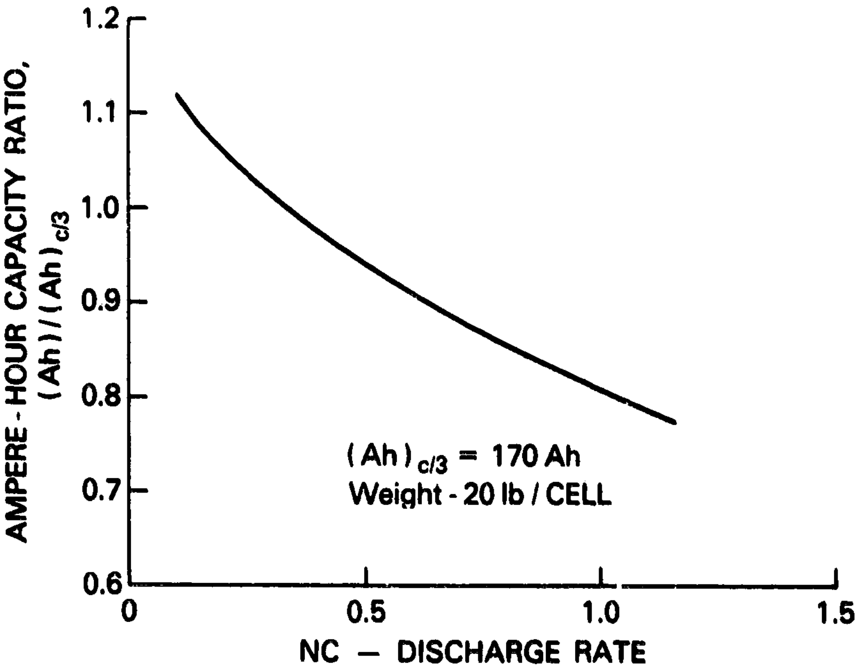

Figure 3-19. Change in $Ah_{Capacity}$ with Discharge Rate for Lead-Acid Batteries

N_m = number of cells in the vehicle battery module

N_s = number of modules in series in vehicle battery pack

N_p = number of parallel strings in the vehicle battery pack.

The voltage droop (E_B/E_o) in the vehicle battery is calculated from Equation (2) using the equivalent power given by Equation (3). The corresponding cell current is given by

$$I_{V/cell} = \frac{(P_B)_V}{N_m\ N_p\ N_s} \Big/ \frac{E_B}{E_o}\ e_o \tag{4}$$

where e_o is the open-circuit voltage of the reference cell, and the battery current and voltage are given by

$$(I_B)_V = N_p\ I_{V/cell} \tag{5}$$

$$(E_B)_V = N_s\ N_m \left(\frac{E_B}{E_o}\right) e_o \tag{6}$$

The updated battery state-of-charge is then

$$S = \frac{\Sigma(I_{V/cell}\ \ \Delta t)}{(Ah)_{V,avg}} \tag{7}$$

where $(Ah)_{V,avg}$ is the vehicle cell Ah-capacity at the average discharge current $(I_B)_{V,avg}$, which is calculated from

$$(I_B)_{V,avg} = \frac{\Sigma\ (I_B)_V\ \Delta t}{t_T} \tag{8}$$

$t_T \equiv$ accumulated driving cycle time

If the battery voltage determined from Equation (6) is less than that required by the motor, then motor voltage and/or power required must by reduced. In some circumstances, the motor voltage can be reduced by a gear shift or, if that is not possible, the power required from the motor can be reduced by load sharing with the heat engine.

Battery behavior during regenerative braking is treated in a manner similar to that described for discharge except that the voltage-current characteristics shown in Figure 3-20 are used for the reference battery.

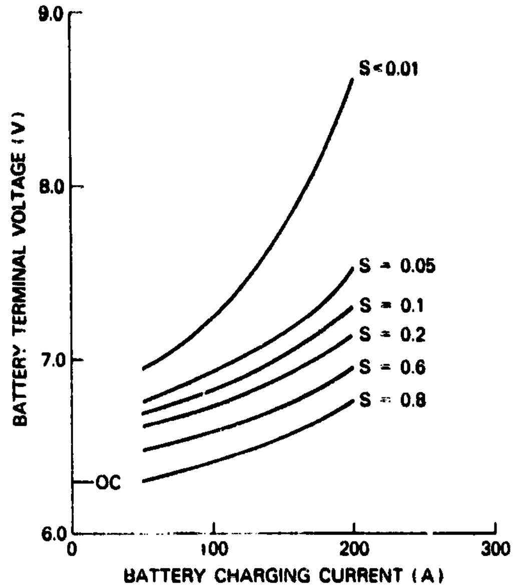

Figure 3-20. Lead-Acid Battery Charging Characteristics[37]

3.4.4.2 Ni-Zn Batteries

The simulation of the behavior of Ni-Zn batteries in the hybrid vehicle is treated in the same manner as discussed in the previous section for lead-acid batteries. The discharge voltage-current characteristics for the reference Ni-Zn battery are shown in Figure 3-21. Those characteristics represent a composite of data obtained from Energy Research Corporation [20] and ESB for developmental batteries/cells. The change in cell capacity (Ah) with average discharge rate for the Ni-Zn batteries is given in Figure 3-22. Due to the lack of charging data for Ni-Zn batteries at charge currents appropriate for regenerative braking, the relative overvoltage required was calculated from the same charging characteristics used for the lead-acid batteries (Figure 3-20).

3.4.4.3 Ni-Fe Batteries

The behavior of Ni-Fe batteries was treated in the same manner as lead-acid and Ni-Zn. The discharge voltage-current and Ah-capacity characteristics used are given in Figures 3-23 and 3-24. These characteristics* were developed from data obtained from Daihatsu Motor Company for the Matsushita Ni-Fe battery which they tested in a subcompact passenger car.

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*Data obtained very recently for Ni-Fe cells being developed by Westinghouse are consistent with Figure 3-17. See the battery supplement for a comparison.

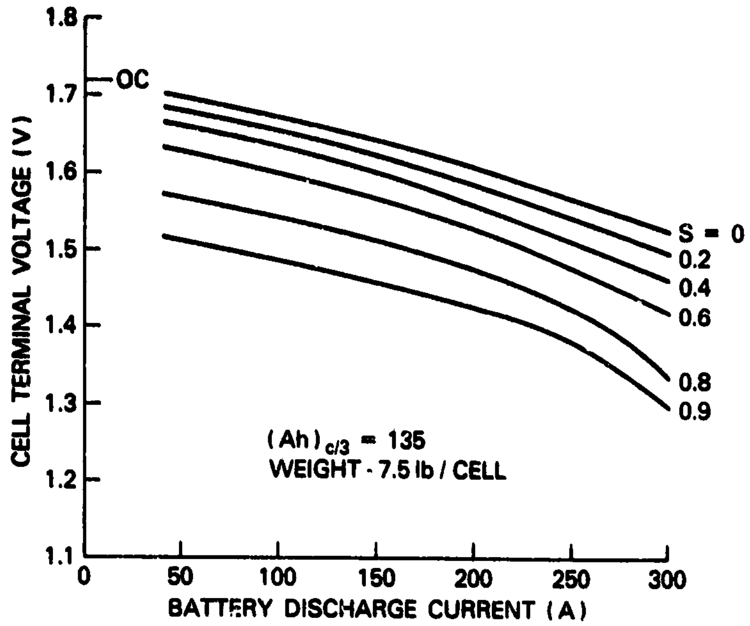


Figure 3-21. Ni-Zn Battery Discharge Characteristics

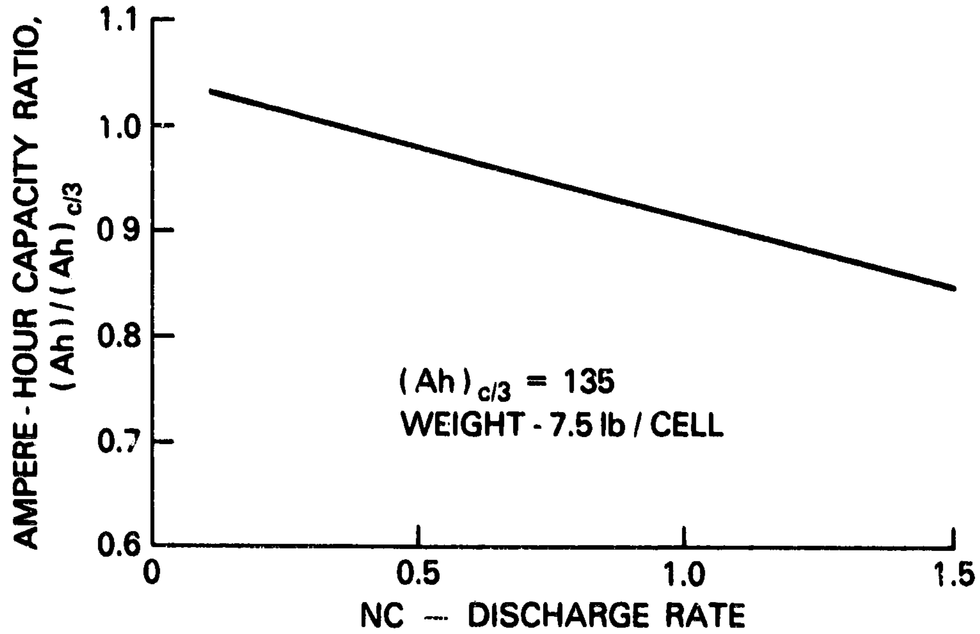


Figure 3-22. Change in $Ah_{Capacity}$ with Discharge Rate for Ni-Zn Batteries

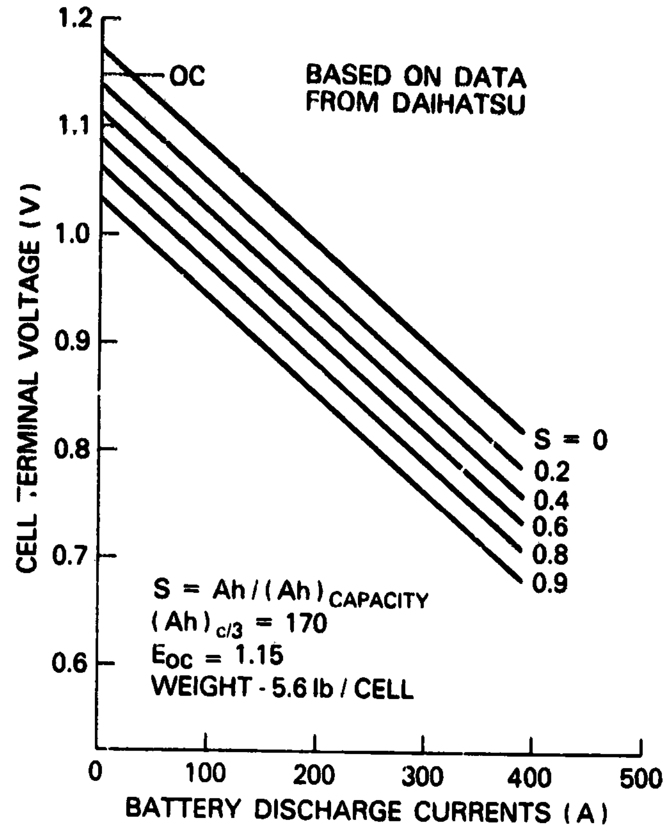


Figure 3-23. Ni-Fe Battery Discharge Characteristics

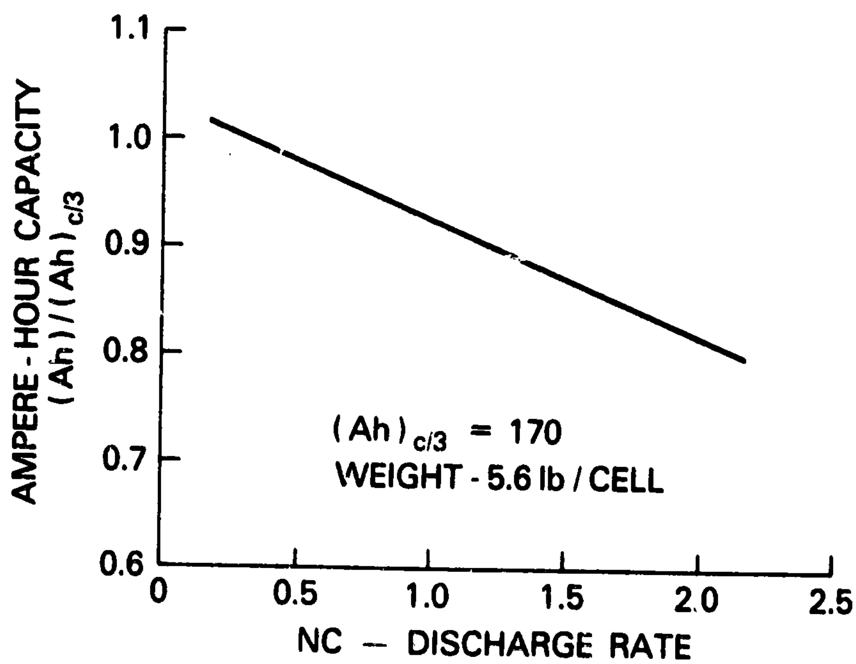


Figure 3-24. Change in $Ah_{Capacity}$ with Discharge Rate for Ni-Fe Batteries

3.5 TRANSMISSIONS, CLUTCHES, AND POWER DIFFERENTIALS

3.5.1 MECHANICAL DRIVE-LINE CONFIGURATIONS

As will be discussed in Section 5, the most attractive hybrid/electric power train configuration is the parallel configuration in which torque from the electric motor and heat engine can be applied to the drive shaft separately or in combination. In addition the drive-line system must accommodate regenerative braking which requires that a negative torque be transmitted to the wheels. As might be expected the mechanical drive-line needed to perform all these functions is relatively complex. This is illustrated in Figure 3-25 which shows a schematic of the parallel hybrid configuration which was studied. As indicated in the figure, there are several options for nearly every component in the system. The decisions as to which options or combination of options is the best depend on a number of factors:

1. Size and weight
2. Drive-line controllability and driveability
3. Maintenance
4. Efficiency
5. Availability and state-of-development
6. Cost

One of the key questions in designing the drive-line system is whether the driver operation in starting from rest after a stop is completely automatic as in the conventional ICE vehicle with an automatic transmission. In that case all the driver does is manipulate the accelerator pedal and the required torque/speed matching is done automatically by the mechanical components (the torque converter and automatically shifted gearbox). The alternative approach is the manual transmission in which the driver manipulates both a clutch and accelerator pedal. As will be discussed later, it does not seem feasible to have the driver determine when to shift gears, as such decisions will be too complex in the hybrid vehicle, but it does seem feasible to use a manually operated clutch to initiate vehicle motion from rest, and, in this way, control the vehicle at low speeds. A number of electric vehicles have been built which use a manual slipping clutch to match the motor and vehicle speeds in initiating motion from rest. It was decided in the present study that the use of the manual clutch did not meet the program goal of having a hybrid vehicle design with mass marketability as a large function of buyers of five-passenger cars in the United States presently prefer automatic transmissions. Hence, all mechanical designs considered in this study operated using only two pedals - an accelerator pedal and a brake pedal. Thus, any clutches would be automatically manipulated. It was, however, deemed satisfactory to have the driver set a multi-position lever for city and highway driving much as some automatic transmissions have positions for low, cruise, and reverse.

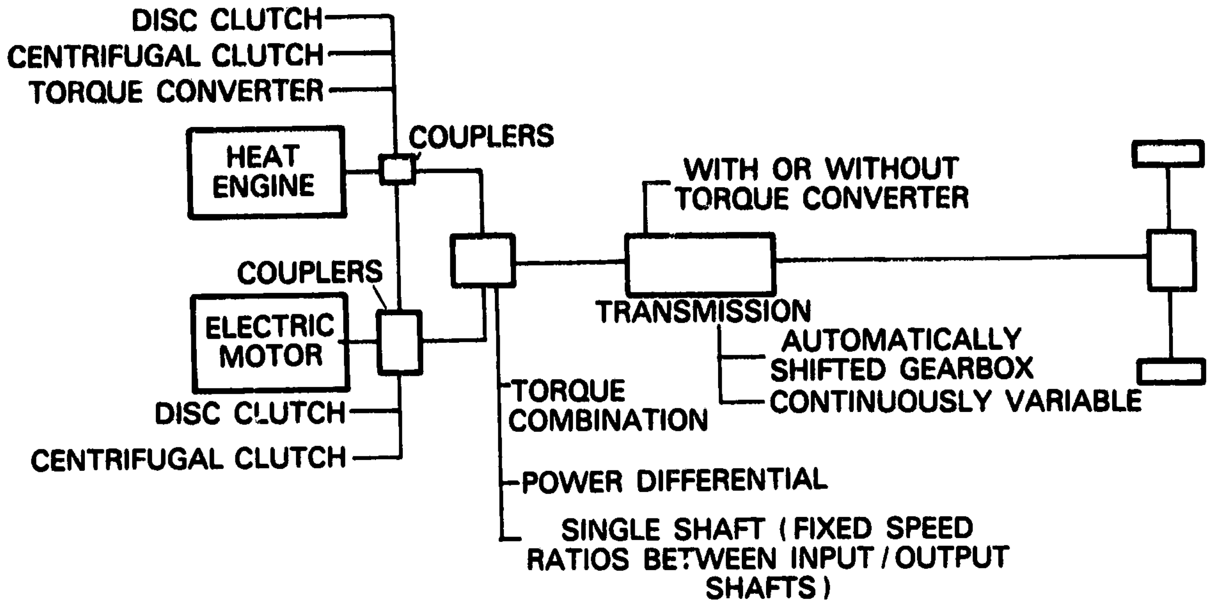


Figure 3-25. Parallel Hybrid Power Train Schematic

With the exception of the continuously variable transmission (CVT), which will be discussed in some detail in Section 3.5.2, all the mechanical components are considered to be production automotive parts even though the exact component arrangements needed for the hybrid/electric vehicle may not presently exist as off-the-shelf items. Packaging of the components for optimum weight and size might also require engineering and development, but this would be the case for any new vehicle (e.g., GM X-body passenger car). Except for possibly the CVT, all the mechanical drive-line components can be mass-produced at acceptable automotive industry costs. Various characteristics of the mechanical components are discussed in the following sections.

3.5.2 TRANSMISSIONS

In all conventional ICE vehicles, a multi-speed transmission is used between the heat engine and the drive shaft to the wheels. The transmission is needed to match engine speed to drive shaft speed over a wide range of vehicle speed. For a parallel hybrid power train, a transmission is needed for the same reason even though the electric drive system can be used to narrow the vehicle speed range over which the heat engine is used. This speed range will depend on battery state-of-charge and widen as the battery is discharged. In an all-electric vehicle, a multi-speed transmission is not necessary between the electric motor and the drive shaft if the electric drive system has armature current control (e.g., a chopper). In that case, the motor speed can be reduced to zero when the vehicle is at rest and the armature control functions as an infinitely variable transmission. Even for all-electric vehicles with armature control, the use of a multi-speed transmission is advantageous for torque multiplication purposes to improve vehicle gradability. Hence, in the hybrid power train, it seems desirable to place the transmission as shown in Figure 3-26 such that both the heat engine and electric motor drive systems benefit from its speed range capability. The shifting logic for the transmission will, however, be dependent on whether the electric or heat engine drive-line are the primary source of power. It should be noted that the electric motors function efficiently at high rpm while heat engines operate most effectively at low rpm and high BMEP.

Two types of transmissions are being considered for the parallel hybrid vehicle. These are the automatically shifted gearbox, similar to that in the standard automatic transmission, and the steel-belt continuously variable transmission (CVT) (Figure 3-27) which was initially developed by Van Doorne in Holland and is currently being further developed in the United States by Borg-Warner. The steel-belt CVT is presently being road-tested in a Ford Fiesta, and it seems likely that such a transmission would be available for the Near-Term Hybrid Vehicle Program. The characteristics of the two transmissions being considered are given in Table 3-25. As noted in the table, the speed range of the two transmissions is similar with the difference that the shifted gearbox achieves

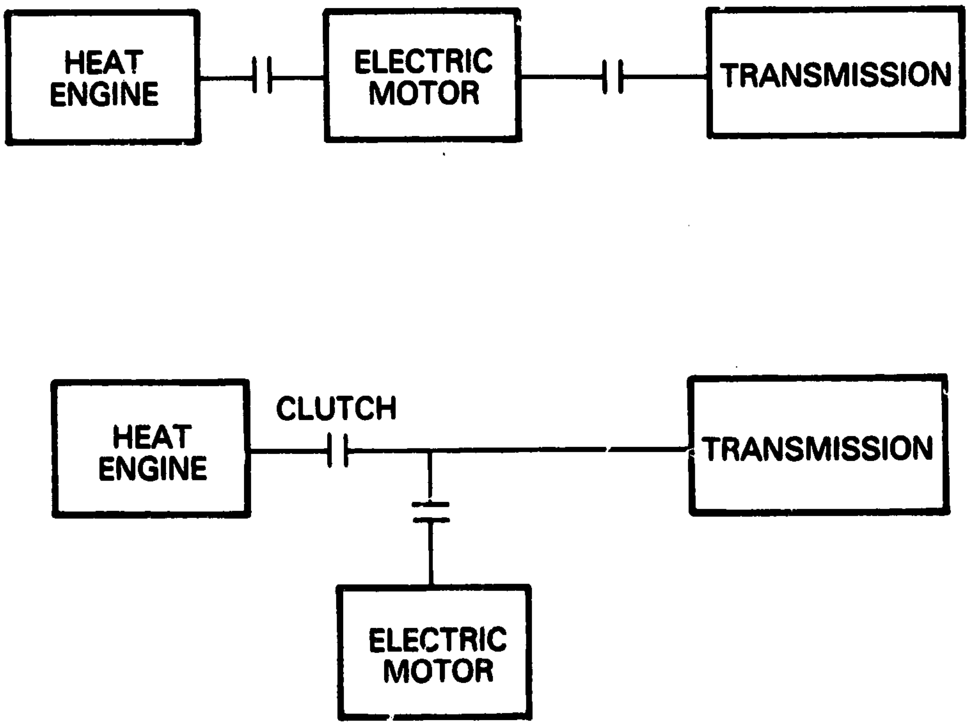


Figure 3-26. Single-Shaft Torque Combining Arrangements

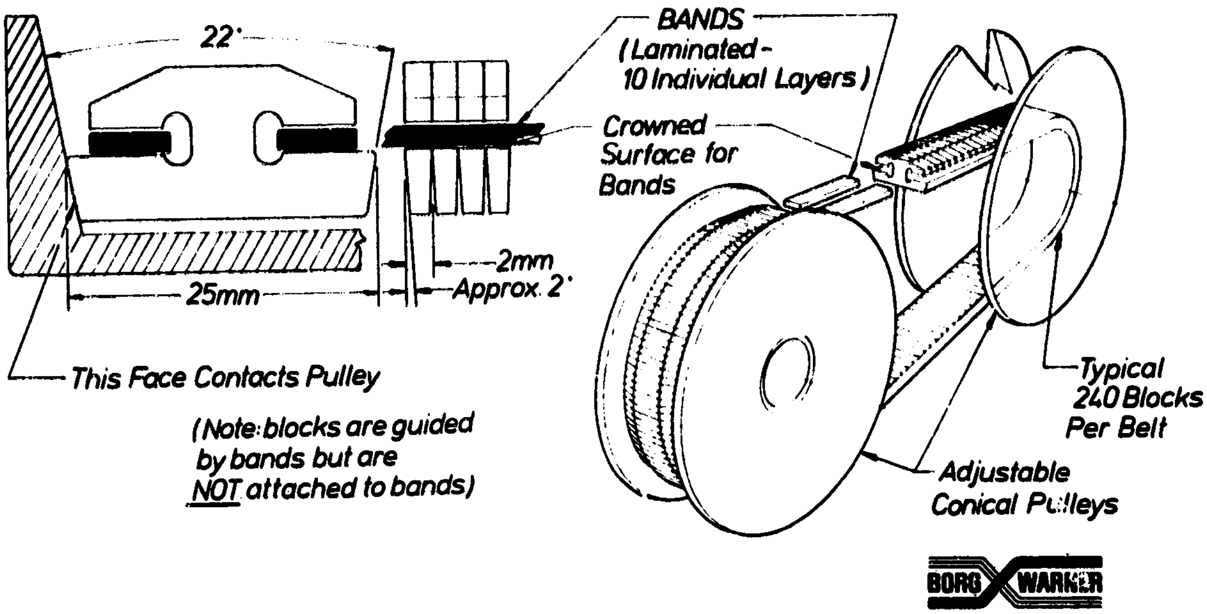


Figure 3-27. Metal Belt Drive

Table 3-25

TRANSMISSION CHARACTERISTICS

<u>Automatically Shifted Gearbox</u>

1st - 3.46:1

2nd - 1.94:1

3rd - 1.29:1

4th - 1:1

<u>Steel Belt Continuously Variable</u>

Overall Speed Ratio: 3.9:1

Speed Reduction Ratio: 2.3:1

Overdrive Ratio: 1.69:1

speed changes in discrete steps and the CVT can achieve the speed changes with infinitely small gradation. This infinitely variable capability of the CVT offers another degree of freedom in system control which is not available with the stepped speed change gearbox. It also should result in a smoother operation of the power train as the vehicle speed is changed. Information available from Borg-Warner indicates that the weight and size of the multi-speed gearbox and the steel-belted CVT are comparable and that one unit can replace the other in a power train design. There is little information available concerning the relative costs of the two transmissions except that Borg-Warner states in Reference 29 that cost of the steel-belt CVT should be comparable to that of the standard three-speed automatic transmission in high-volume production.

A critical factor in comparing the shifted gearbox and CVT transmissions is their efficiency in transmitting mechanical power at various input speeds, speed ratios, and torque levels. In the past much of the potential advantage of the CVT has been lost by its lower efficiency at low torque (power) levels. Hence, it was of particular interest to evaluate the relative efficiency of the automatically shifted gearbox and the steel-belted CVT. Both transmissions have losses due to internal rotating friction and the necessity to provide hydraulic power to operate/control (shift or vary the transmission geometry) them. It is advantageous to compare the transmissions in terms of losses rather than efficiency. This is done in Figure 3-28 for input speeds up to 5000 rpm using available test data. The CVT data was obtained from Borg-Warner, and the automatically shifted gearbox data was obtained from Triad Services. The shifted gearbox tested by Triad was specially modified for use in an electric vehicle without a torque converter. It appears from Figure 3-28 that the two transmissions have essentially the same losses with those of the automatically shifted gearbox being slightly higher, except possibly at very high rpm. The differences in the losses do not appear to be significant, but this certainly requires additional attention during the Preliminary Design Task.

To date, all the hybrid vehicle simulation calculations using HYVEC have been made for a four-speed automatically shifted gearbox with a gear ratio range of 3.5:1 (the standard four-speed VW gearbox). More detailed transmission studies will be undertaken in the Preliminary Design Task to investigate the improvements in vehicle performance and energy efficiency that would result from using the steel-belted CVT having a speed ratio range of up to five to one. More detailed transmission loss data will be used than has been included to date.

3.5.3 CLUTCHES AND FLUID COUPLERS

As indicated in Figure 3-25 clutches or fluid couplers are needed at numerous places in the hybrid/electric power train. The clutches are used to connect/disconnect the heat engine and electric motor into the power train and to modulate the torque applied during periods of torque blending and vehicle acceleration from rest. There are three coupling devices that can be used:

1. Flat-disc pressure clutch
2. Centrifugal clutch
3. Torque converter

All three coupling devices have a slipping action with the degree of self-control or regulation increasing from 1 to 2 to 3. The prime factors in selecting the coupling device to be used in the drive-line are ease of operation and control and efficiency (associated energy losses during coupling/uncoupling and after coupling has been achieved). The clutches are more efficient and the torque converter easier to operate and control.

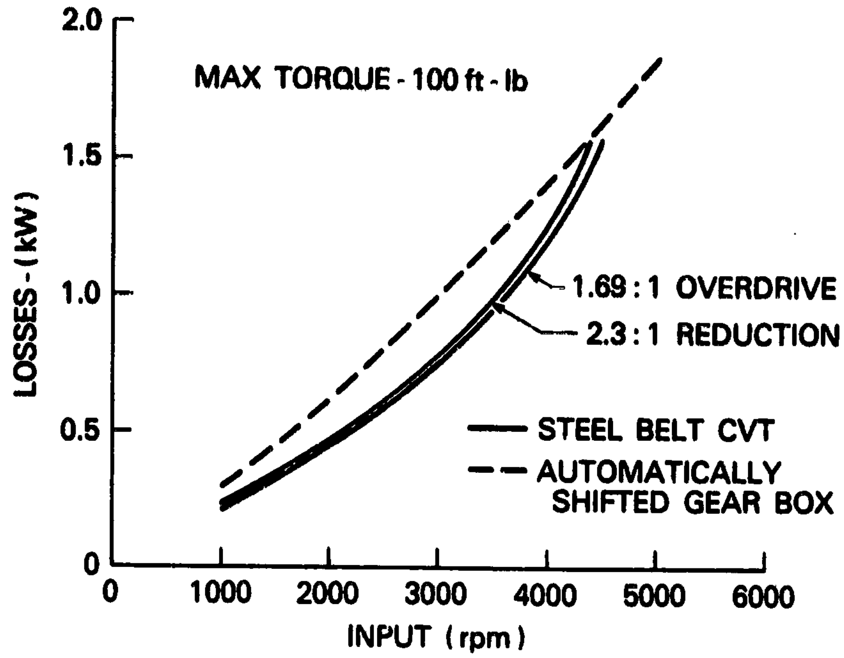


Figure 3-28. Losses for a Steel-Belt CVT and Automatically Shifted Gearbox as a Function of Input Speed

First consider the torque converter, which could be used either to couple the heat engine into the power train or to couple the hybrid output shaft to the transmission. In the latter position, the torque converter must handle negative torque during regenerative braking. Torque converters are a key component in all automatic transmissions used in conventional ICE vehicles. Hence, their characteristics are well-known in both the driving mode in which power (positive torque) is transferred to the transmission from the engine and in the coasting mode in which engine braking (negative motoring torque) is applied to the vehicle. In the driving mode the torque converter has a torque multiplication and in the coasting mode the torque converter acts as a simple fluid coupling with a torque ratio of unity. The characteristics of a 10-inch automotive torque converter (200 ft-lb torque capacity) are given in Figures 3-29, 3-30, as a function of speed ratio ($SR = N_o/Ni$). The key characteristic of the torque converter is the capacity factor $K (= N_i/\sqrt{T_i})$, which relates input speed N_i to input torque T_i for a given speed ratio SR. The torque converter must operate along the K versus SR line with the resulting efficiency and torque ratio TR indicated in Figures 3-29, 3-30. Torque converters are not particularly attractive for use in hybrid/electric power trains to couple into the transmission for several reasons. First, they are lossy during accelerations and even beyond the coupling point, the efficiency is only 85 to 95%. Efficiency is more important in a hybrid/electric vehicle than for an ICE vehicle because of its effect on battery requirements. The efficiency of the torque converter can be improved using lockup. Secondly, without armature control, it is difficult to reduce the idle rpm of the dc separately excited electric motor to values compatible with the torque converter. The idle losses of the torque converter are intolerable unless the motor idle rpm is significantly less than

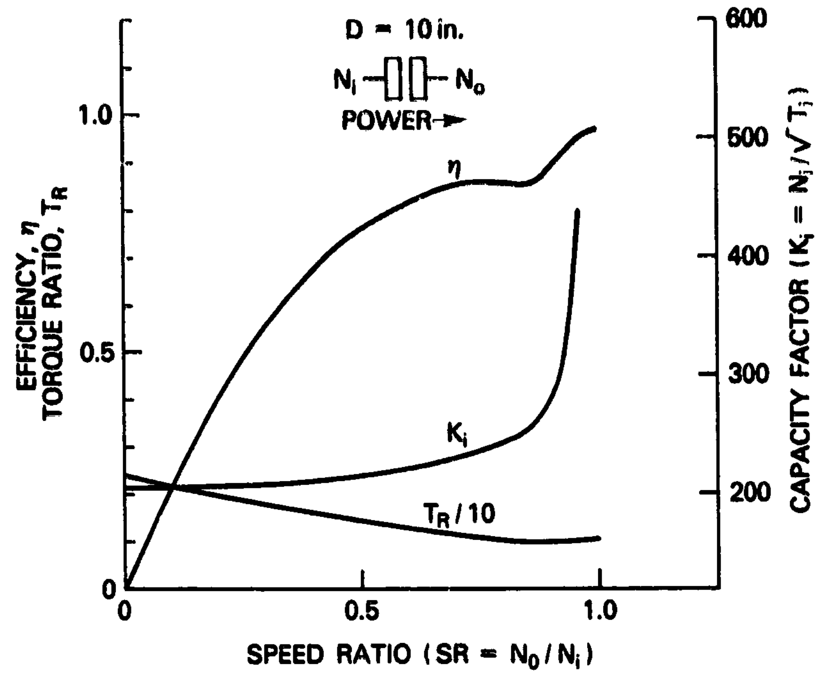


Figure 3-29. Torque Converter Characteristics for Driving

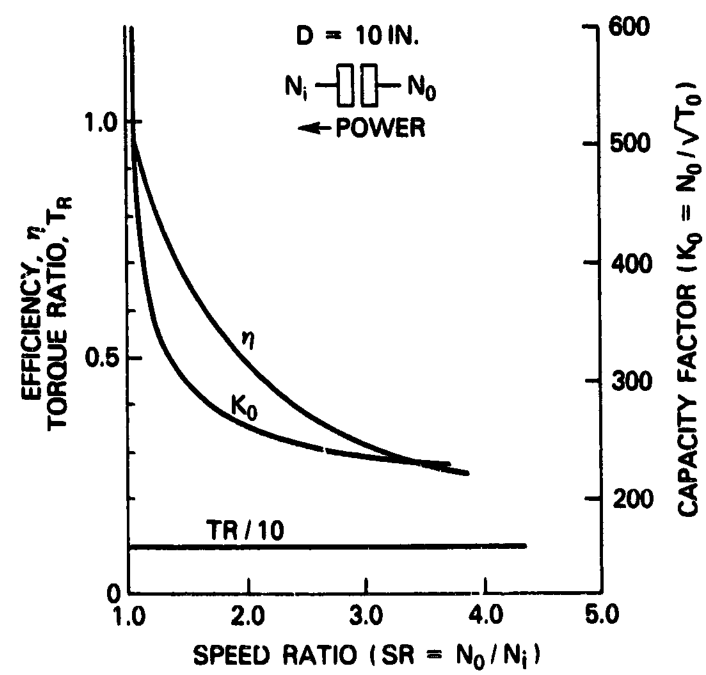


Figure 3-30. Torque Converter Characteristics for Regenerative Braking

1000 rpm. Even with battery switching this is not easy. Thirdly, in the regenerative braking mode, the speed reduction across the fluid coupling increases and the efficiency decreases as the vehicle speed decreases for a fixed braking torque. Neither of these characteristics is conducive to good regenerative braking. The torque converter does not seem attractive for coupling the heat engine to the power train. In this application the torque converter would be used primarily as a means of smoothing the introduction of engine torque when it is needed to assist the electric motor. Associated with this smoothing is the "windup time" needed before the torque converter can transmit engine torque. In addition, the relatively low efficiency of the torque converter will reduce the highway fuel economy of the hybrid vehicle by five to ten percent unless lockup is used. The torque converter requires a larger hydraulic pump than the shifting gearbox alone and this further increases the accessory loads and losses. Hence, in the present study, the intent is to use clutches rather than fluid coupling elements in the hybrid power train.

In principle, clutches (either pressure plate or centrifugal) are simple, efficient devices. The basic problems associated with their use in the hybrid vehicle are ones of modulation and control and the resultant effect on vehicle driveability, especially at low speeds and in close quarters, such as in heavy traffic, garages, and parking lots. Another problem that should not be overlooked is that of holding the vehicle on a grade. All of these problems become more difficult to handle when the possibility of using a manually-operated clutch to couple the hybrid power train to the transmission is excluded from consideration. This means that all clutch systems must be automatically actuated and modulated using the single accelerator pedal and/or inputs from the microprocessor. Choice of the best clutch type for coupling the heat engine and electric motor to the drive-line is complicated by the number of different functions and modes the clutch must accommodate. For example, the same clutch that must couple the electric motor to the drive-line to initiate vehicle motion from rest must also remain coupled when the motor/generator is being driven by the heat engine to recharge the batteries on the road. There has been considerable effort by Fiat [23] and by Borg-Warner in connection with the CVT development to use centrifugal clutches which close at a selected rotational speed. Such clutches could be used in the hybrid power train if detailed study of the various operating modes show they can work properly under all required conditions. Otherwise, modulated pressure plate clutches will be used. Such clutches are the most difficult to control (open/close smoothly) but are clearly the most flexible in use. The effect of clutch slip (losses) is not presently included in HYVEC, but this can be done without great difficulty. This will be done as part of the development of the clutch modulating strategy.

3.5.4 POWER DIFFERENTIAL

In the parallel hybrid drive train, provision must be made for combining the output torques of the heat engine and electric motor when both are operating and for permitting each to power the vehicle alone when that is desired. Of particular importance is the blending of the two outputs when the operating strategy changes from the use of one of the prime movers to the use of both of them. Another important consideration is that the gearing between the heat engine and electric motor permit each to operate at speeds and loads at which they are most efficient. The simplest gearing arrangement is to maintain a fixed ratio between the rpm of the heat

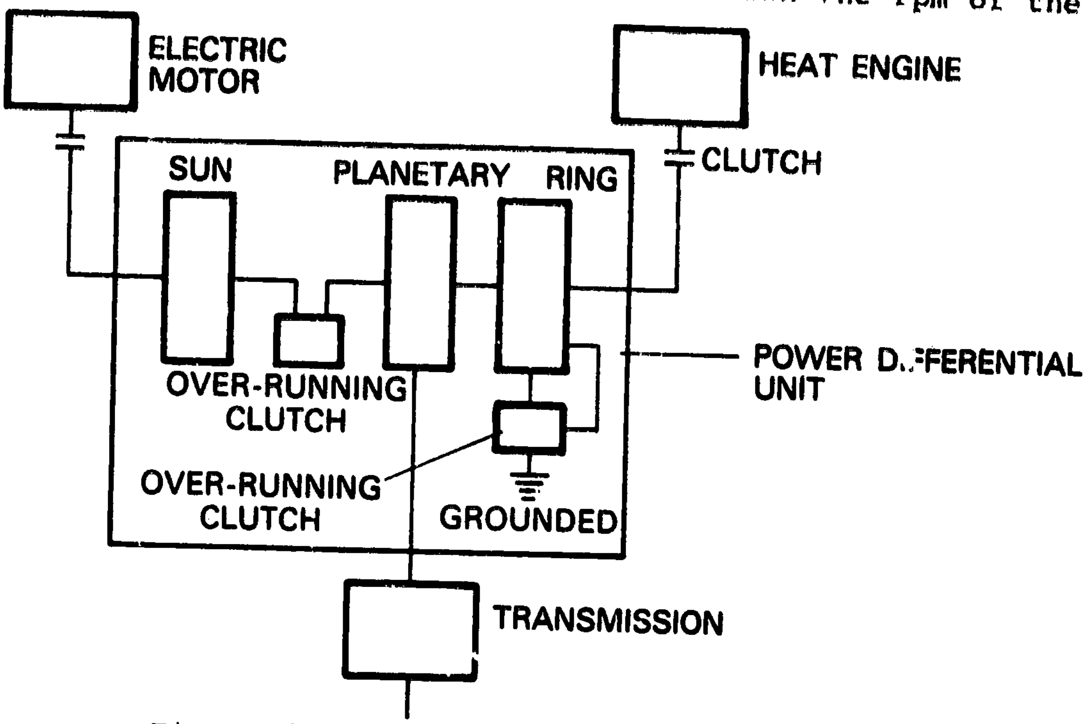


Figure 3-31. Schematic of the Power Differential Arrangement

engine and electric motor. This arrangement is often referred to as the single-shaft configuration (Figure 3-26) even though it does not require that both prime movers are actually on the same shaft. The single-shaft arrangement is the easiest to analyze and control because the motor and heat engine speeds are dependent only on the vehicle speed and transmission gear, independent of motor or heat engine output. The motor/engine speeds are the same for a given vehicle speed and gear regardless of whether only one of both the motor and engine are powered. There are two considerations which make the simple single-shaft arrangement less than optimum. First, fixing the speed ratio between the heat engine and electric motor may not permit both to operate near their optimum efficiency for a given vehicle speed and desired load sharing. Secondly, it may prove difficult to modulate the coupling/uncoupling clutches such that the power blending is done with the desired smoothmess (lack of jerk) and vehicle driveability.

An alternative approach for combining the heat engine and electric motor torques is shown in Figure 3-31. This will be referred to as the "power differential arrangement." There are a number of ways in which the differential could be connected, but the one shown in Figure 3-31 seems to be the most advantageous for the present application. In the arrangement shown, the electric motor is connected to the sun gear, the heat engine to the ring gear, and the drive shaft (input to the transmission) to the planetary. Note that provision is made for driving the vehicle using the electric motor alone by reacting the engine shaft torque to a grounded over-running clutch. Similarly, there is an over-running clutch between the sun gear and planetary shafts which lock the two shafts together when only the heat engine is powered. This second over-running clutch also maintains the electric motor shaft at high rpm - thus, in a ready condition when the motor/generator is needed for power or braking. It is felt that the power differential arrangements will permit smoother blending of the heat engine and electric motor torques because the inherent characteristic of the differential is that the sum of the torques must be zero. As the heat engine or electric motor is brought on power to assist the other, the speed ratios must automatically change to accommodate the inherent torque summing relationship. In addition, the differential arrangement has another degree of freedom not available in the fixed speed ratio single-shaft arrangement, and this may permit more efficient operation of the components especially when coupled with a CVT.

The added complexity of the power differential arrangement is evident. Control of the system is also surely more difficult than the single-shaft arrangement. The governing relationships between the loads and speeds in the power differential have been derived, but not as yet implemented in HYVEC. All the hybrid vehicle simulation calculations made to date have been for the single-shaft arrangement. Further evaluation of the need for the more complex power differential will be made during the Preliminary Design Task. As discussed in Section 6, the vehicle layouts have been made leaving space for the power differential in the event it is found to be either necessary or highly desirable from a driveability or system efficiency point-of-view.

3.6 MICROPROCESSORS

The need for and the use of microprocessors in the control system for the hybrid power train have been recognized from the outset of the Design Trade-Off Studies (Task 2). All the component trade-off studies and control strategy investigations were performed assuming that complex and rapid decisions/calculations could be made automatically on-board the vehicle utilizing microprocessors. Otherwise the load sharing between the heat engine and electric motor (which is so critical to the satisfactory operation of the hybrid vehicle) would not be possible. Considerable work has been done in recent years in the application of microprocessors to automotive systems - both in conventional ICE vehicles[23,24,25] and in all-electric vehicles.[26,27] That experience is the basis for the confidence that microprocessors can be used in the hybrid vehicle control system with a minimum of difficulty and without the development of new technology. By the

1982 to 1985 time period of the Near-Term Hybrid Vehicle Program, it is expected that the use of microprocessors for engine control in ICE vehicles will be commonplace.

First consider the use of microprocessors in conventional ICE vehicles. As discussed in Reference 24, much activity has occurred and is occurring in the field of automotive electronics. Most of that work has been done in connection with the development of programmed engine controls needed to meet the stringent exhaust emission standards now in effect with a minimum fuel economy penalty. Probably the most advanced engine control system currently being marketed is the L-Jetronic electronic fuel-injection system produced by Bosch(28) which is used in a number of European and Japanese sports cars (e.g., Datsun 280-Z). Other electronically controlled engine systems are under development by various auto companies for use with three-way catalysts.(25) Hence, both microprocessors and the associated sensors/instrumentation which can operate in the automotive/heat engine environment are available for use in the hybrid vehicle program. In addition, as the use of automotive microprocessors becomes more common, the cost of such devices will decrease and their reliability will increase significantly.

The GE/DOE Near-Term Electric Vehicle utilizes a microprocessor to control the electric drive system. The development of the microprocessor for that application is described in detail in References 26, 27. The microprocessor system for the electric drive in the hybrid vehicle will be much like that developed for the General Electric all-electric vehicle. Hence, most of that effort is directly applicable to the present program.

As presently envisioned, the hybrid vehicle control system will utilize three microprocessors - one for the heat engine, one for the electric drive system, and a system microprocessor which will oversee the entire power train/vehicle operation. Detailed consideration of the microprocessors was not included in the Design Trade-Off Studies after it was determined that microprocessor technology presently available would meet the requirements of the hybrid vehicle application. The detailed microprocessor study is now being done as part of the Preliminary Design (Task 3).

The cost of the microprocessor was determined from the cost/producibility study done in connection with the Near-Term Electric Vehicle Program.(14) The microprocessor cost was included in the controller cost expressed as $/kW because the microprocessor is designed as an integral part of the controller circuitry. The best estimate that could be made of microprocessor cost separate from that of the armature and field choppers was $100 - $150 OEM depending on volume of production. This would result in a contribution of $3 to $4/kW (based on peak electric drive power output) to the total controller cost. This value of microprocessor specific cost ($/kW) was used to determine the total controller cost for the different dc and ac systems.

Section 4

POWER TRAIN CONFIGURATION CLASSIFICATION AND DEFINITION OF TERMS

Section 4

POWER TRAIN CONFIGURATION CLASSIFICATION AND DEFINITION OF TERMS

4.1 INTRODUCTION

Before proceeding to an evaluation and comparison of the various hybrid/electric power train configurations, it is appropriate to discuss the classification of the various configurations and to define the terms used.

4.2 POWER TRAIN CLASSES

In the present study four classes of power train configurations were defined:

1. All-electric
2. Series hybrid
3. Parallel hybrid
4. Conventional ICE

The distinguishing characteristics of the power train clases are the following:

- All-electric. All the torque to the drive shaft is supplied by an electric motor which draws electrical energy from the battery.
- Series hybrid. All the torque to the drive shaft is supplied by an electric motor which draws electrical energy from the battery and a heat engine-driven generator.
- Parallel hybrid. The torque to the drive shaft is supplied by an electric motor and heat engine combination. The electric motor draws electrical energy from the battery, but the heat engine can recharge the battery using the motor in the generator mode.

In all the hybrid/electric power trains defined, electrical energy is stored in a battery pack. The electric range of the vehicle depends primarily on the kWh storage capacity of the battery pack which is termed the primary storage unit. Since, as discussed in Section 3.3, batteries are power-limited, there is a maximum power that can be supplied to the electric motor from a battery pack of a given weight. Hence, for a specified vehicle electric range and power-to-weight ratio, the size of the battery pack may be determined by power rather than range requirements. This results in a considerably higher vehicle weight than would be the case if only the range requirement had to be met. One approach to satisfying the power and range requirements without increasing the size of the battery pack is to incorporate secondary energy storage into the all-electric or hybrid power trains. The secondary storage unit must have a very high power density so that it can supply the needed additional power without much additional weight. The prime candidate for the secondary energy storage unit is the flywheel. The power rating of the flywheel unit is determined by the rating of the continuously variable transmission (CVT) used to connect it to the power train. The size of the flywheel is determined by its energy density (Wh/lb) and the time for which the flywheel unit is to supply the power boost to the drive-line system.

Schematics (block diagrams) for each of the classes of hybrid/electric power trains are given in Figures 4-1 through 4-3. Diagrams are shown for power trains having only primary storage, as well as ones having both primary and secondary storage units. The vehicle synthesis results discussed in Section 5 of this report were obtained by analyzing power trains arranged in exactly the manner diagrammed in Figures 4-1 through 4-3. Other arrangements are possible with the transmission and CVT placed in other positions in the drive-line, but the arrangements presented in the figures permitted a self-consistent comparison of the various drive-line configurations.

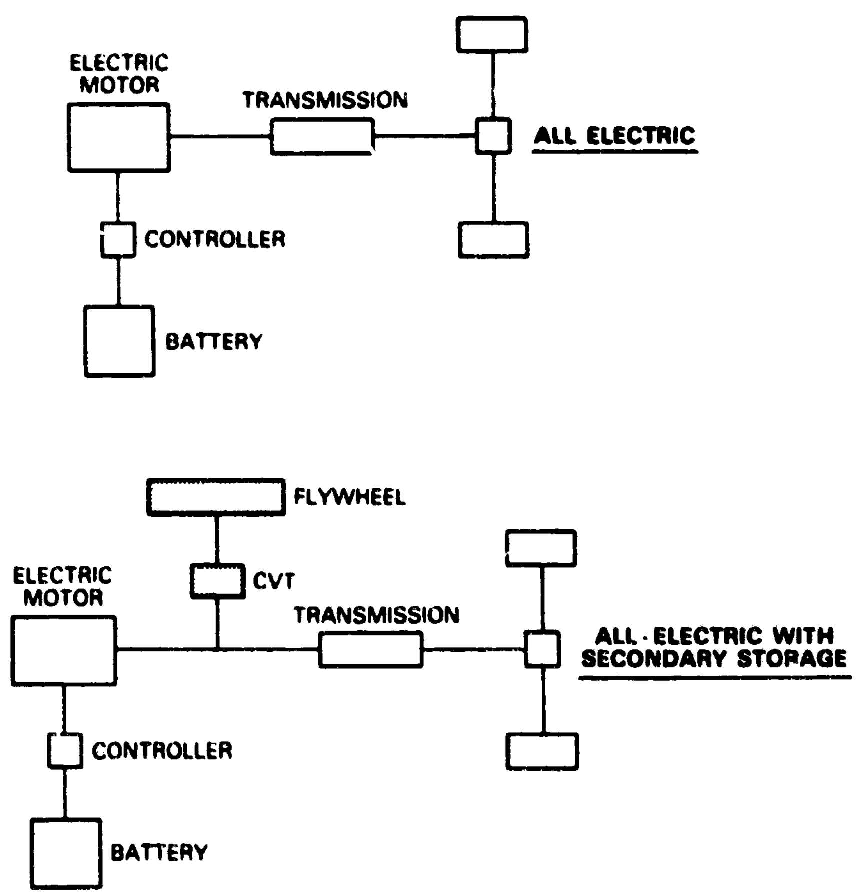


Figure 4-1. Electric Vehicle Power Train Schematics.

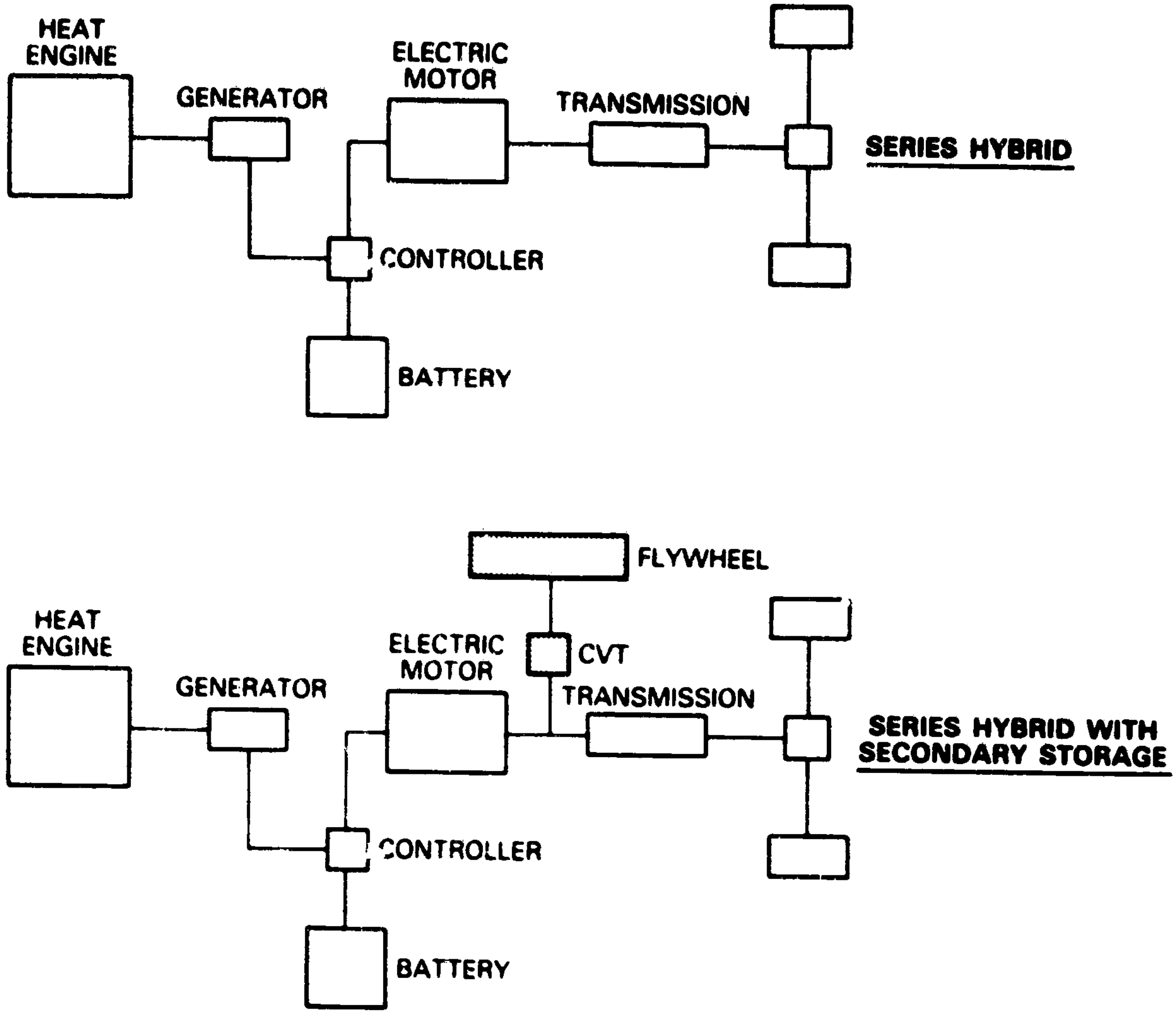
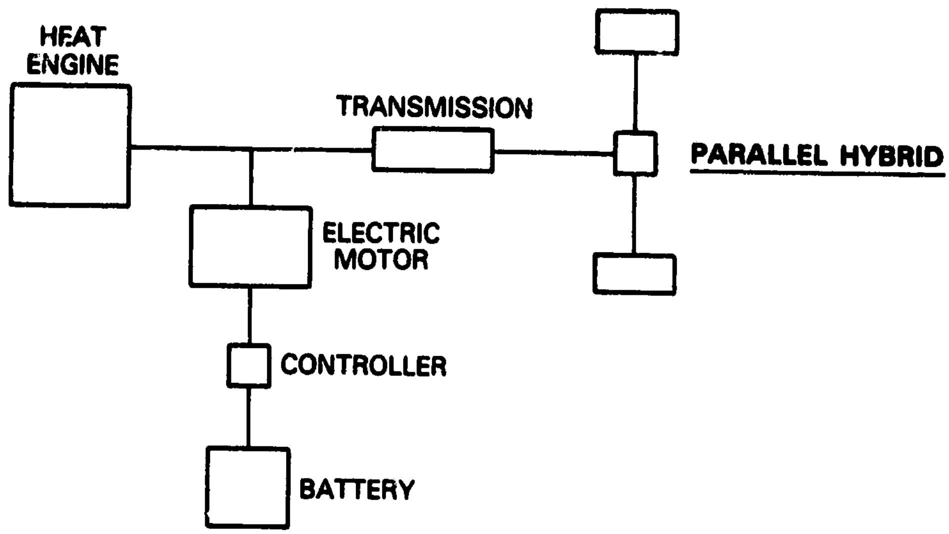


Figure 4-2. Series Hybrid Power Train Schematics



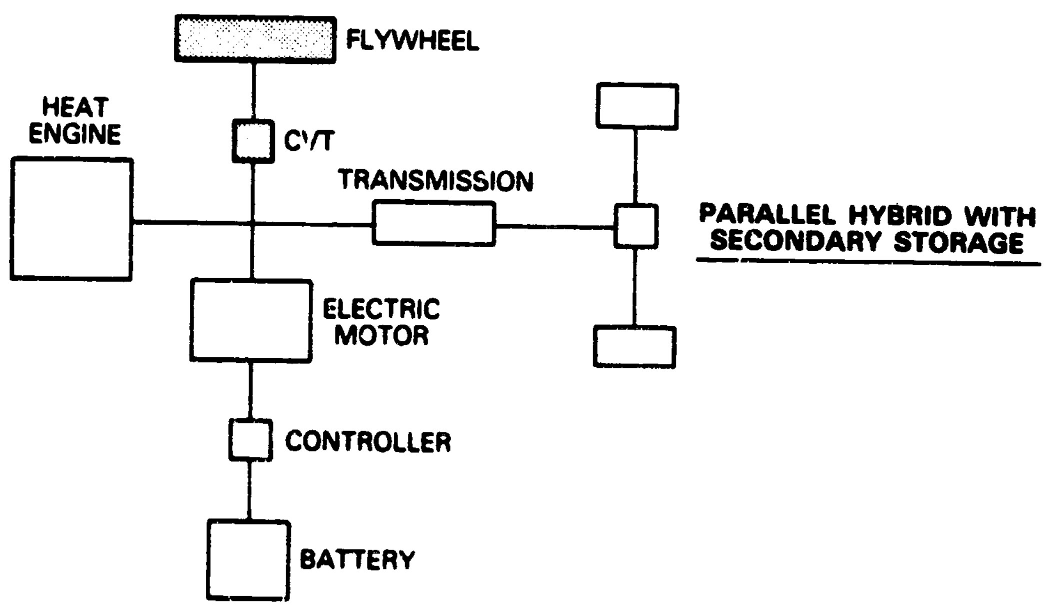


Figure 4-3. Parallel Hybrid Power Train Schematics

Section 5

EVALUATION AND COMPARISON OF CANDIDATE POWER TRAINS BY USE OF VEHICLE SYNTHESIS AND ECONOMICS

Section 5

EVALUATION AND COMPARISON OF CANDIDATE POWER TRAINS BY USE OF VEHICLE SYNTHESIS AND ECONOMICS

5.1 INTRODUCTION

The design trade-off results discussed in this section were obtained using the vehicle synthesis and economics computer program (HYVELD). A complete listing of the program is given in Volume III. Each vehicle/power train combination studied is described in terms of a number of input parameters which characterize the vehicle, its performance and use, and the power train configuration and each component in it. The input parameters required for a typical HYVELD calculation are given in Table 5-1. The input parameters are grouped into three categories:

1. Hybrid/electric vehicle design characteristics
2. Economic factors
3. Conventional vehicle characteristics

The hybrid/electric vehicle design characteristics concerned with drive-line components are based on the results of Section 3, and those concerned with vehicle performance and use are based on the Mission Analysis Test Report, SRD-79-010. The economic factors are based primarily on the Jet Propulsion Laboratory's Assumptions and Guidelines Package, New-Term Hybrid Passenger Vehicle - Phase I. The characterization of the conventional ICE reference vehicle is based on the vehicle characterization results given in the Mission Analysis Report (Sections 3 and 5).

Before discussing the design trade-off results, it is of interest to note the major design trade-offs which were considered and the criteria used in comparing and evaluating the various drive-line configurations and components in this initial screening of hybrid/electric powertrain possibilities. The major design trade-offs considered were the following:

1. Parallel versus series configurations
2. Secondary energy storage (e.g., flywheels)
3. Heat engine/electric drive-line power split
4. Battery type
5. Heat engine type
6. Electric drive type (e.g., dc or ac)

Each of the design trade-offs was examined for a range of vehicle power-to-weight ratios and design electric range values. The primary criteria of comparison and evaluation were the following vehicle parameters:

Table 5-1

TYPICAL INPUT FOR A HYVELD CALCULATION HYBRID/ELECTRIC VEHICLE PARAMETERS

Power-to-Weight Ratio (kW/lb)	0.020
Specific Weight Engine (lb/kW)	5.000
Specific Weight Transmission (lb/kW)	1.200
Specific Cost Engine ($/kW)	8.500
Specific Cost Transmission ($/kW)	2.500
Fuel Economy in City (mi/gal)	22.000
Fuel Economy on Highway (mi/gal)	32.000
Fuel Economy (composite)	26.000
Conventional Vehicle Cost ($)	5700.000
Conventional Vehicle Lifetime (yr)	10.000
Chassis Weight (lbs)	2150.000
Payload Weight (lbs)	300.000
Power-to-Weight Ratio	0.020
Base-Loaded Vehicle (lbs)	2450.000
Range (design)-Miles: Pure Storage	75.000
Range (design) Miles: Hybrid City Driving	35.000
Electric Drive-Line Efficiency	0.800
Mechanical Drive Efficiency	0.900
Power Plant Efficiency	0.350
Cost of Additional Chassis Weight ($/lb)	0.620
Weight Propagation Factor	0.300
Miles Traveled per Year	11852.000
Fraction of Miles in City	[illegible]
Fraction of Miles in City, Electric	[illegible]
Energy Consumption in City (kWh/ton-mile)	[illegible]
Energy Consumption on Highway (kWh/ton-mile)	[illegible]
Fraction of Energy from Engine in City	[illegible]
Fraction of Energy from Engine on Highway	[illegible]
[illegible] ($/kWh)	[illegible]
Gasoline Price ($/gal)	[illegible]
Specific Weight of Motor Generator (lb/kW)	[illegible]
Specific Weight of [illegible] (lb/kW)	[illegible]
Specific Weight of [illegible] (lb/kW)	[illegible]
Specific Cost of Battery [illegible] ($/kWh)	[illegible]
[illegible]	[illegible]
[illegible] ($/kW)	[illegible]
[illegible] ($/kW)	[illegible]
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[illegible]	[illegible]
[illegible]	[illegible]

- Total weight (lb)
- Selling price ($)
- Operating cost (¢/mi)
- Annual gasoline savings (gal/yr)
- Net dollar savings ($/yr)

For the most part, the vehicle use and economic factors were held constant during the design trade-off calculations. Variations in those factors were treated in the Sensitivity Analysis Task which is examined in a separate report.[31]

5.2 POWER TRAIN CONFIGURATIONS

Vehicle synthesis calculations were made for a number of general power train configurations in five-passenger vehicles including all-electric, parallel hybrid, series hybrid, and conventional ICE. The use of secondary energy storage (e.g., flywheel) was considered in the all-electric and hybrid power trains. The results of the calculations form the basis for the comparisons given in the following subsections of parallel versus series configurations and powertrains with and without secondary energy storage.

5.2.1 PARALLEL VERSUS SERIES CONFIGURATIONS

Total vehicle weight and selling price for vehicles utilizing parallel and series configurations are shown in Figures 5-1 and 5-2. Also shown in the figures are the corresponding values for all-electric and ICE vehicles. Several points are clear from the figures. First, the weight and cost of vehicles using a series configuration are in all cases higher than those for vehicles of the same performance using a parallel power train. Secondly, the differences become much larger as the power-to-weight ratio of the vehicle is increased. For vehicles having low performance (Kp = 0.012), the differences in vehicle weight and selling price are probably manageable, but for vehicles having performance comparable to a conventional ICE vehicle (Kp _ 0.02), the differences in weight and cost become sufficiently large that the use of a series hybrid configuration should not be considered. The effect of the heat engine power fraction (F_{HE}) on the weight and cost of hybrid vehicles (both series and parallel configurations) is given in Figures 5-3 and 5-4. The effect of F_{HE} is relatively small at Kp = 0.02 but becomes greater at higher performance levels.

The weight and cost of the series hybrid are high for several reasons:

1. All the traction power must be applied by the electric motor resulting in a relatively heavy and high-cost motor.

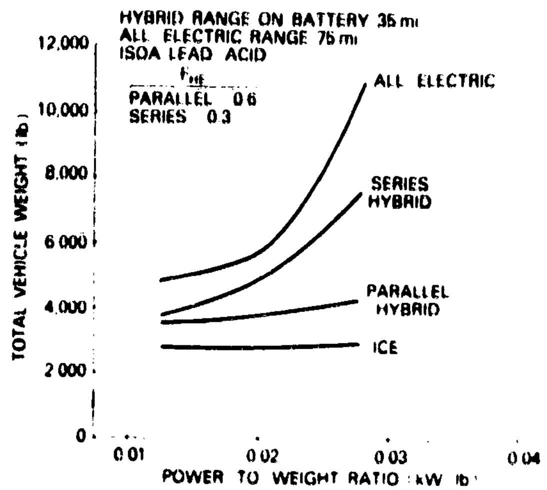


Figure 5-1. Effect of Power Train Configuration on Total Vehicle Weight

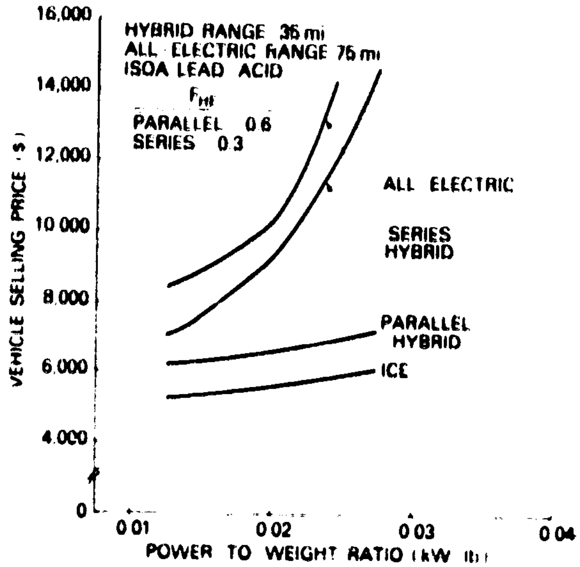


Figure 5-2. Effect of Power Train Configuration on Selling Price

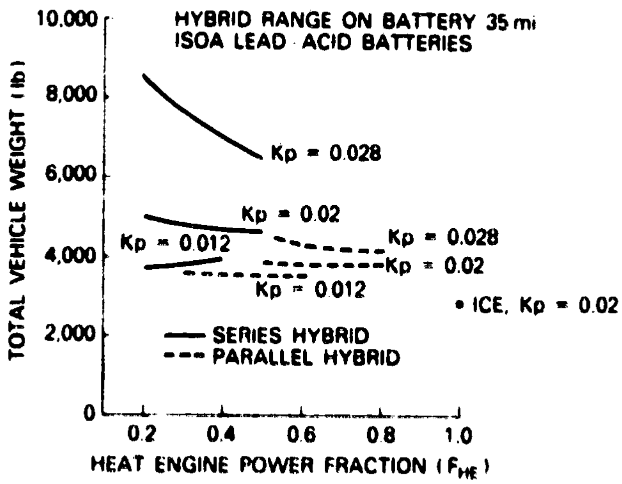


Figure 5-3. Effect of Heat Engine Power Fraction on Total Vehicle Weight for Series and Parallel Hybrid Vehicle

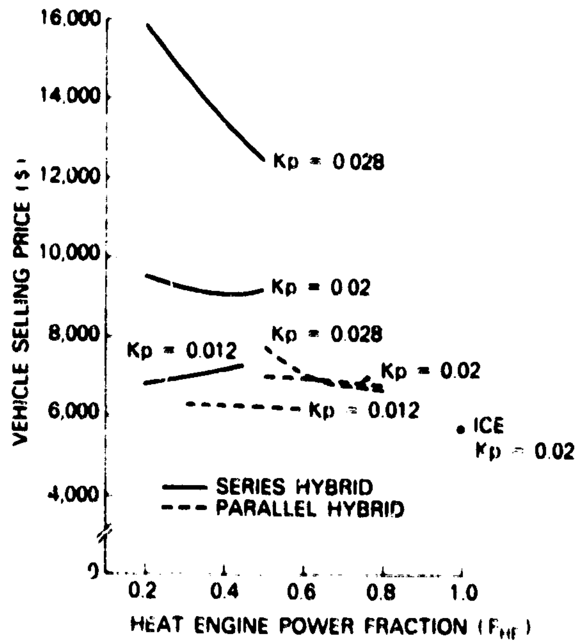


Figure 5-4. Effect of Heat Engine Power Fraction on Vehicle Selling Price for Series and Parallel Hybrid Vehicles

2. The battery pack is sized by power requirements rather than range unless the design range of the hybrid vehicle is relatively large.

3. A separate generator is required to supply electric power to the motor.

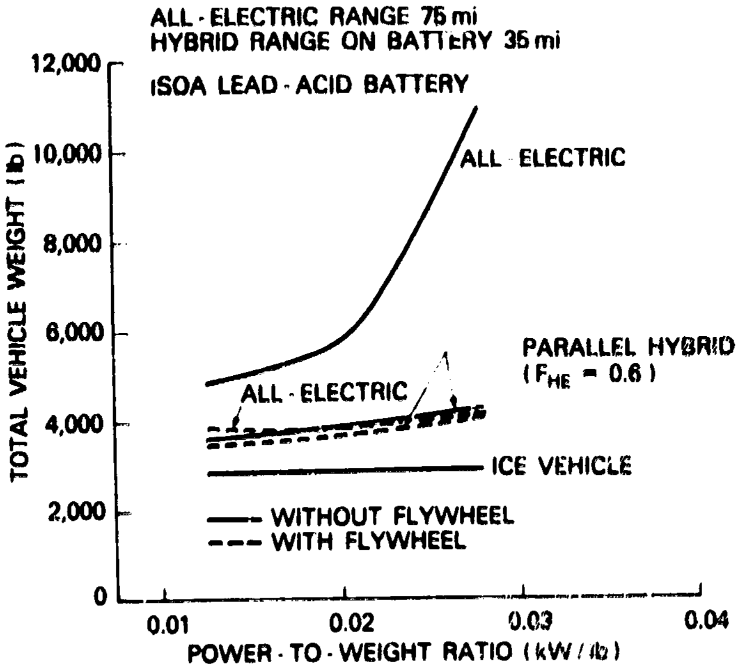
In addition, the conversion of a significant fraction of the total power from mechanical to electrical (with or without storage in the battery) and then back to mechanical to drive the vehicle is inherently inefficient. This latter consideration is not included in the vehicle synthesis calculations but must be considered in comparing parallel and series configurations for applications (low performance vehicles) in which the vehicle weight and selling price differences are not large.

Based on the calculations discussed in this section, no further consideration will be given to series configurations in this study.

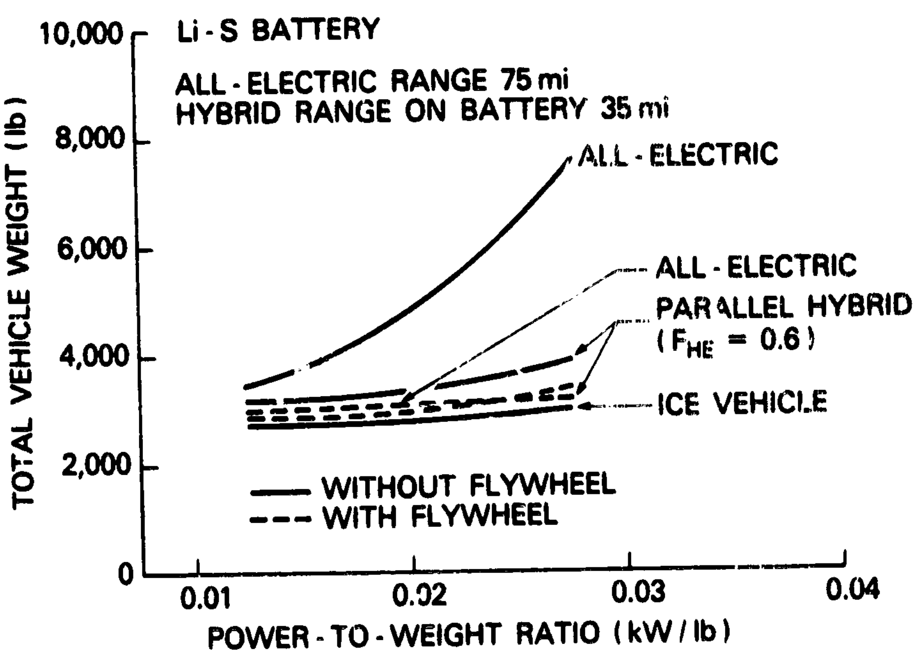
5.2.2 SECONDARY ENERGY STORAGE

Vehicle synthesis calculations were also made for powertrains utilizing secondary energy storage, such as flywheels or high-power density batteries, to supplement the power capability of the primary battery pack. Secondary storage has a large effect on vehicle weight and cost in those cases in which the battery pack is sized by peak power requirements rather than range. This is much more likely to be the case for an all-electric vehicle or a series hybrid than for a parallel hybrid in which the heat engine provides at least one-half the peak power. This is illustrated in Figures 5-5a and 5-5b where vehicle weight is shown for various driveline configurations for power-to-weight ratios between 0.012 and 0.028. The dramatic reduction in vehicle weight for the all-electric drive using a flywheel is particularly noteworthy. The calculations indicate that for a parallel hybrid vehicle using lead-acid batteries, the weight advantage of using a flywheel is small at best. However, if Li-S batteries become available at some future time, strong consideration of the use of secondary storage (either high power density lead-acid batteries or a flywheel) in a parallel hybrid would appear to be warranted.

Since the calculations have shown that for relatively high power density batteries, such as lead-acid, Ni-Zn, secondary energy storage does not have a significant advantage for parallel hybrid vehicles, secondary energy storage was not considered further in this study. It was concluded, however, that for high-performance all-electric vehicles or parallel hybrid vehicles using batteries with high energy density and relatively low power density (e.g., Li-S), secondary energy storage is likely to show significant advantages.



(a)



(b)

Figure 5-5. Effect of Flywheel Energy Storage on the Weight of Hybrid and All Electric Vehicles

5.3 HEAT ENGINE/ELECTRIC DRIVE POWER SPLIT

One of the key considerations in designing a parallel hybrid/electric vehicle is the power split between the heat engine and the electric drive systems. In the present study, the power split is expressed in terms of the parameter, F_{HE}, which is the fraction of the peak power which can be supplied by the heat engine. The fraction which can be supplied by the electric drive, F_{DE}, is simply $1-F_{HE}$. Calculations were made using the HYVELD program for various combinations of vehicle power-to-weight ratio (Kp) and heat engine power fraction (F_{HE}). The results for vehicle weight are given in Figure 5-6 for several types of batteries. In general, the vehicle weight increases with power-to-weight ratio and decreases with increasing F_{HE}. Marked decreases in weight with F_{HE} for a given battery type indicate that the battery is sized by peak power requirements and increasing F_{HE} has significantly reduced the weight of batteries required.

Figure 5-6 indicates that selection of the engine power fraction depends both on power-to-weight ratio and battery type. For lead-acid and Ni-Zn batteries, an F_{HE} equal to about 0.6 results in near-minimum weight especially for Kp = 0.02. It must also be noted that unless the absolute power rating of the electric drive system is sufficiently large to permit vehicle operation primarily on electricity in most urban driving, the gasoline saved using the hybrid vehicles will be unacceptably small. Hence, the

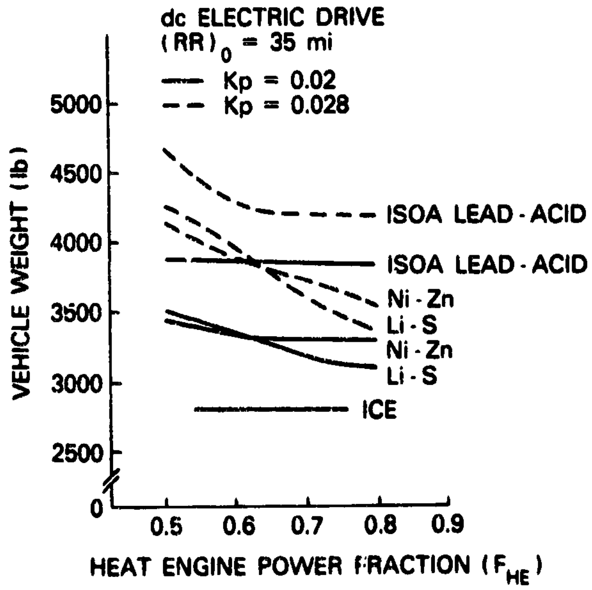


Figure 5-6. Effect of Heat Engine Power Fraction on Vehicle Weight for Various Types of Batteries

general approach in selecting F_{HE} for a specified Kp is to fix the absolute power rating of the electric drive system at that required for most urban driving and to determine the heat engine size required to satisfy the remaining power requirements. Using this approach, the optimum F_{HE} will increase with Kp.

5.4 BATTERY TYPE

Extensive calculations were made for parallel hybrid vehicle designs using various battery types. The batteries considered were the following.

- ISOA lead-acid
- Advanced lead-acid
- Ni-Zn
- Ni-Fe
- Li-S

The characteristics assumed for each of the battery types are given in Table 5-2. The same battery characteristics were used in all the vehicle synthesis calculations.

Vehicle weight and selling price are given in Figures 5-7, 5-8, 5-9, and 5-10 as a function of power-to-weight ratio and design electric range for all the battery types. The results shown are for a parallel hybrid power train and a dc electric drive. As would be expected, the advanced batteries (all types other than ISOA lead-acid) lead to hybrid vehicle designs having lighter weight and, for the most part, a lower selling price. In terms of weight, the reductions are significant, but not so large as to make the use of ISOA lead-acid batteries unfeasible in the hybrid vehicle. In terms of selling price and operating cost (as discussed later), hybrid vehicles using ISOA lead-acid are projected to have only slightly higher values than those using the more advanced batteries. Hence, it is concluded that the advanced batteries certainly would have an advantage if and when they are developed, but that those battery developments are not required to meet the design and economic goals of the Near-Term Hybrid Vehicle Program. This important point will be made several additional times in the continuing discussion of the various battery types.

It is of interest to consider the effect of battery type on vehicle operating cost. In this consideration, the following vehicle design parameters are fixed: Kp = 0.02, $(RR)_O$ = 35 mi, F_{HE} = 0.6. These values are felt to be near optimum for a parallel hybrid vehicle meeting the Jet Propulsion Laboratory's design specifications. Vehicle selling price and operating cost are shown in Figure 5-11 for the various battery types. Results are shown for a low-cost dc electric drive system without armature control (battery switching plus field control) in high-volume production (greater than one million units per year). This

Table 5-2

STORAGE UNIT CHARACTERISTICS USED IN THE DESIGN CALCULATIONS

Storage Unit Battery Type	Primary Storage: Without Load Leveling			Primary Storage: With Load Leveling		Secondary Storage		Cost		Cycle Life (a)	Lifetime (a)	Energy Exponent (b)	Battery Salvage Cost Credit (%)
	Wh/lb	W/lb (c)	W/lb (d)	Wh/lb	W/lb (c)	Wh/lb	W/lb	$/kWh	$/lb	Number of Cycles	Years		
Lead-Acid [illegible]	[illegible]	[illegible]	[illegible]	20	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	800	[illegible]	[illegible].26	15
Lead-Acid ADV	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5	[illegible].26	[illegible]
Ni-Zn	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3	[illegible]	[illegible]
Ni-Fe	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	6	[illegible]	[illegible]
[illegible] ADV	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5	[illegible]	10
Flywheel	--	--	--	--	--	7	N.A.	[illegible]	2.[illegible]	10^6	10	N.A.	N.A.

a [illegible] as lifetime in years.

b $\frac{Wh/lb}{(Wh/lb)_o} = \left(\frac{R}{R_o}\right)^a$

a = energy exponent

R_o = [illegible] vehicle design range

R = calculated vehicle range

$(Wh/lb)_o$ = value given in table

c [illegible] second pulse

d [illegible] second pulse

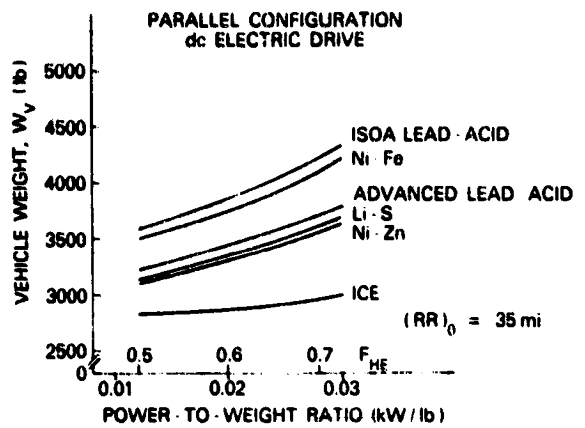


Figure 5-7. Effect of Power-to-Power Weight Ratio on Hybrid Vehicle Weight for Various Types of Batteries

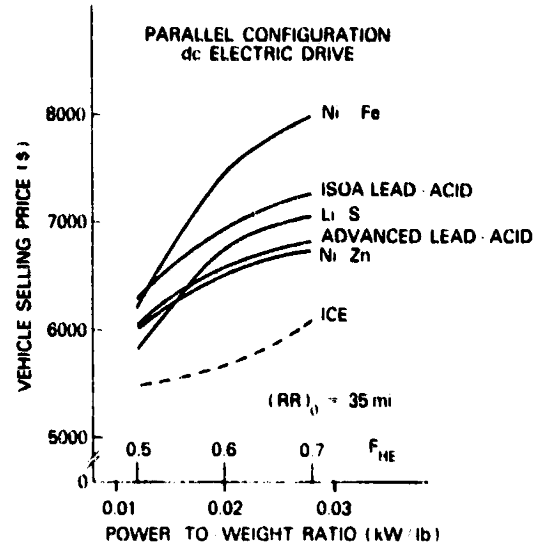


Figure 5-8. The Effect of Power-to-Weight Ratio on the Hybrid Vehicle Selling Price for Various Types of Batteries

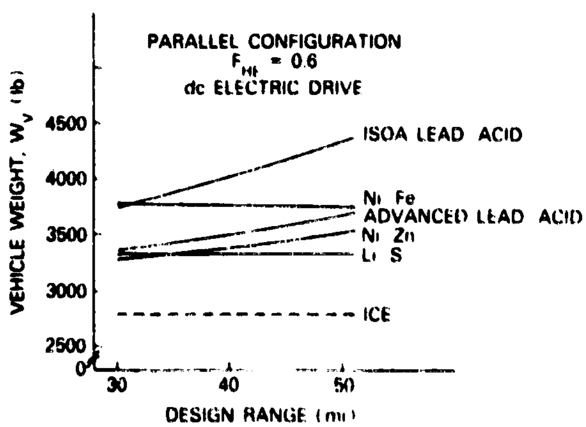


Figure 5-9. Effect of Design Electric Range on Hybrid Vehicle Weight for Various Types of Batteries

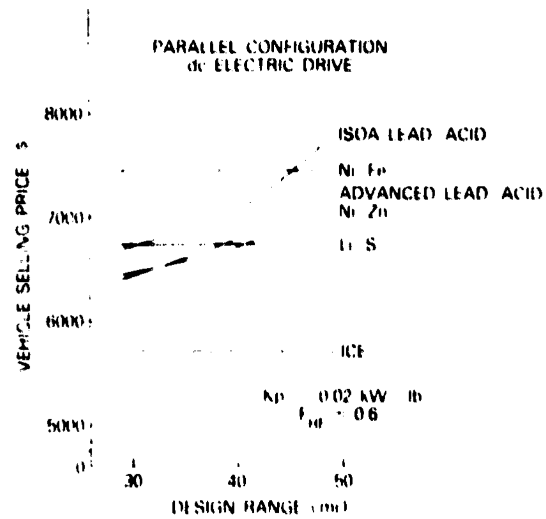


Figure 5-10. Effect of Design Electric Range on Hybrid Vehicle Selling Price for Various Types of Batteries

and not battery type per se. In addition, the hybrid vehicle must be designed and built to exploit the inherent long life of the electrical drive components so that the rate of depreciation and resale value of the hybrid will be superior to that of the ICE vehicle.

Another criterion for comparing the attractiveness of hybrid vehicles using different battery types is the amount of gasoline saved per year. This is shown in Figure 5-12. Note that there is not a large variation between the battery types with the gasoline savings ranging between 258 gal/yr using ISOA lead-acid and 290 gal/yr using Li-S. This reflects the relatively small differences in vehicle weight for the different batteries. The reference ICE vehicle would use 480 gal/yr. Hence, the gasoline savings range between 54 and 60%. As indicated in Table 5-3, most of the gasoline saving occurs in urban driving where the fraction saved ranged between 72 and 75%. On the highway the gasoline saved is small (percent saved ranges from 0 to 13%). The effect of battery type on total energy use (fuel plus electricity including the electrical power plant conversion efficiency) is shown in Table 5-3. All the battery types show a net energy saving, with hybrid vehicle using ISOA lead-acid batteries having a 19.5% saving and those using the advanced lead-acid and Li-S batteries having about a 28% saving.

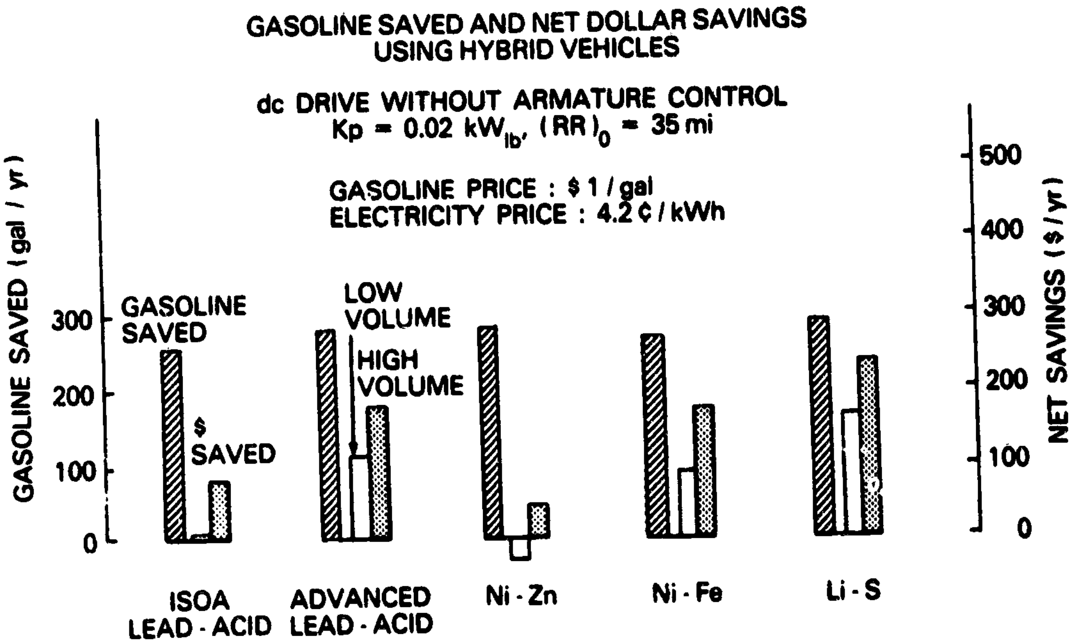


Figure 5-12. Gasoline Saved and Net Dollar Savings Using Hybrid Vehicles

An important factor affecting market penetration of hybrid vehicles is the annual net cost saving to the purchaser of such a vehicle compared to purchase and operation of the reference ICE vehicle. The net cost saving is a strong function of the price of gasoline (or diesel fuel). This is shown in Figure 5-13 for

Table 5-3

EFFECT OF BATTERY TYPE ON ENERGY USE IN A HYBRID VEHICLE

Battery Type	Change in Energy Use (kWh/mi)		Energy Use (kWh/mi)		Energy Saved (%) Composite (65/35)
	Urban	Highway	Urban	Highway	
ISOA Lead Acid	-0.574	+0.079	1.086	1.219	19.5
Advanced Lead Acid	-0.693	-0.054	0.967	1.086	28.3
Ni-Zn	-0.695	-0.056	0.965	1.084	28.5
Ni-Fe	-0.522	+0.138	1.138	1.278	15.7
LI-S	-0.681	-0.041	0.979	1.1	27.5
ICE Reference Vehicle 22 mi/gas - Urban 32 mi/gal - Highway			1.66	1.14	

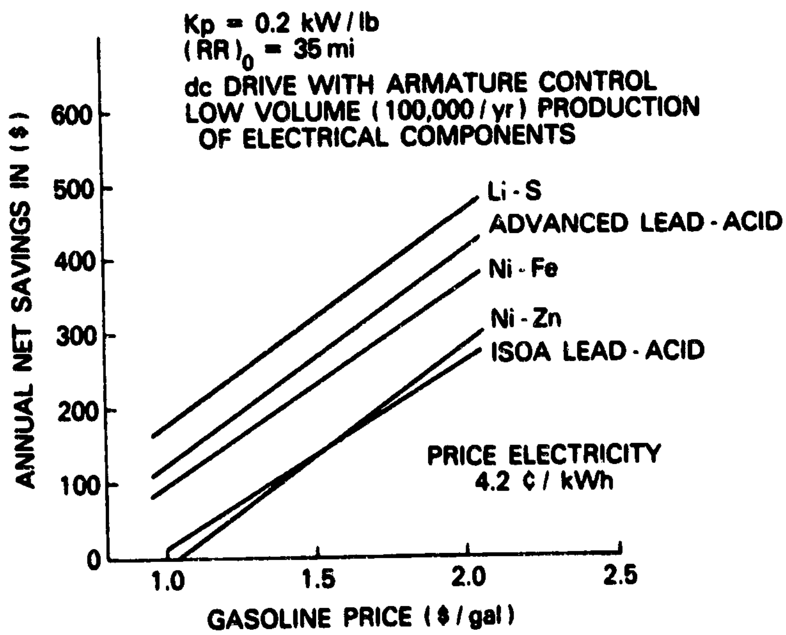


Figure 5-13. Annual Net Dollar Savings as a Function of Gas Price

the various battery types using the low-cost dc electric drive (without armature control electronics). Results are shown in Figure 5-14 for both low-volume (100,000 units/yr) and high-volume (10^6 units/yr) production rates of the electric components (e.g., motors, controller, batteries, etc.). It should be noted that the annual net savings become quite attractive between $1-$2/gal for the fuel. For ISOA lead-acid and Ni-Zn batteries, the

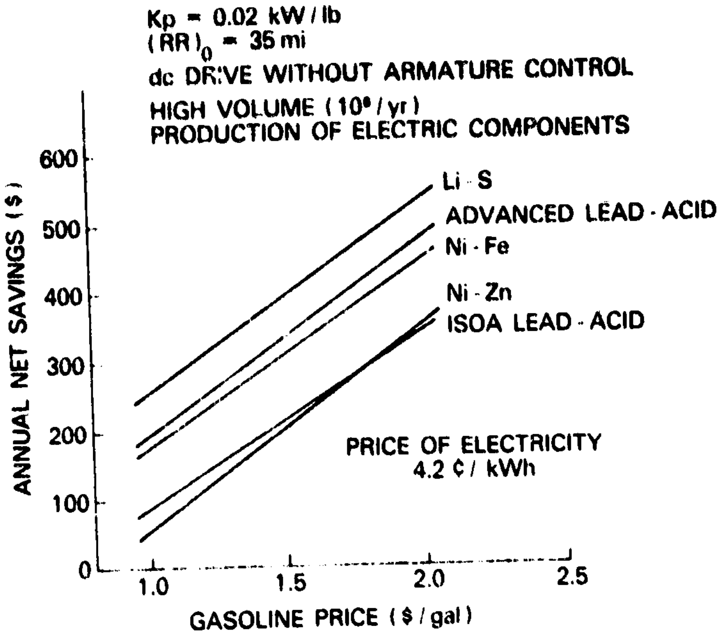


Figure 5-14. Annual Net Dollar Savings as a Function of Gas Price

projected annual savings in the relatively near-term are \$100-\$200, and for the other batteries, they are much higher. Note that these savings are those beyond the break-even point with the hybrid vehicle.

5.5 ENGINE TYPE

Vehicle synthesis calculations were made using various types of heat engines. The primary objective of these calculations was to determine the effect of engine type on hybrid vehicle weight and selling price. The results of the calculations are shown in Figure 5-15 compared with the corresponding values for the reference ICE vehicle. The effect of engine type on hybrid vehicle weight is relatively small with the gasoline rotary engine yielding a vehicle weight of 3800 lb and the Stirling engine yielding a vehicle weight of 4100 lb. The difference in vehicle selling price are more significant with the extremes being \$7000 with the rotary engine and \$7760 with the Stirling engine. The vehicle weight and selling price with the conventional ICE are 3890 lb and \$7017, respectively.

As discussed in considerable detail in Section 3.1, the choice of heat engine for the hybrid vehicle application depends on many factors in addition to weight and cost. A key consideration is engine availability and state-of-development. The heat engine used in the hybrid vehicle program must be readily available and require minimum modification for the special hybrid application because engine development is both time-consuming and expensive. This means that the gas turbine and Stirling engines, which are both in the early stages of development for passenger cars (especially in the 50-70-hp range), could not be considered for the

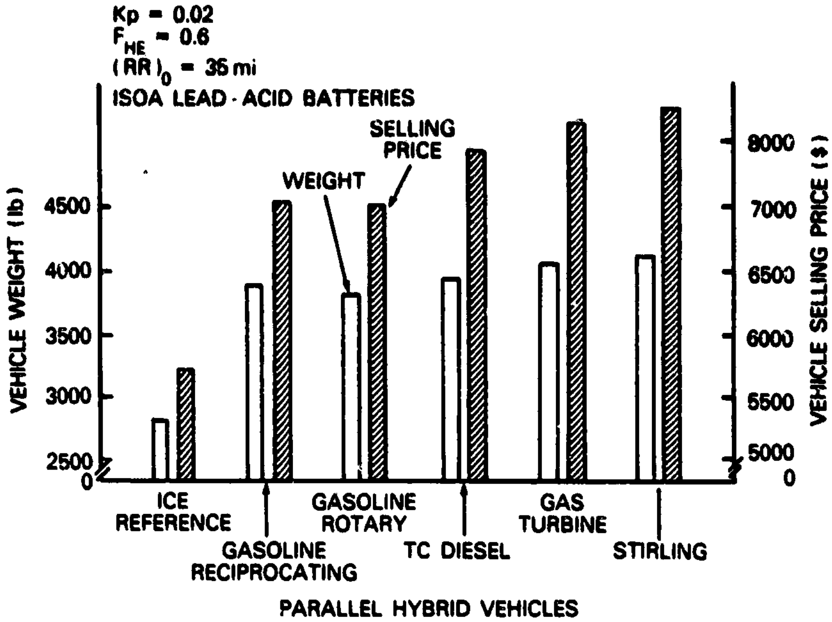


Figure 5-15. Comparison of the Weight and Selling Price of Parallel Hybrid Vehicles

Near-Term Hybrid Vehicle (NTHV) program even if they were attractive in other respects.

There is little doubt that from the fuel economy and emissions points-of-view the Stirling engine would be very attractive (Reference 32) if its high potential in those areas can be demonstrated by future developmental work. In addition, the Stirling engine is a multi-fuel engine, and in this regard it is also attractive for future applications. Unfortunately, it is likely that the inherent bulk and weight of the Stirling engine would preclude its use in a hybrid vehicle even if current development programs prove successful. Also, the continuous character of the combustion process in the Stirling engine is likely to preclude on/off engine operation.

The only gas turbine engine which has even been suggested for use in the NTHV program is the Williams WR 34 single-shaft engine which has been developed for use with a high-speed generator. This engine in its present state of development could only be considered for use in a series hybrid because the controls needed to operate the engine over a wide range of speed and power in a parallel hybrid have not been developed. In addition, little has been done to date concerning emissions from the WR 34. Vehicle synthesis calculations were made to compare vehicle designs using the gas turbine and gasoline engines. The results are shown in Figures 5-16, 5-17, and as expected, the weight and cost penalties associated with the series hybrid are significant and made serious consideration of the use of the Williams WR 34 engine in the hybrid program unattractive.

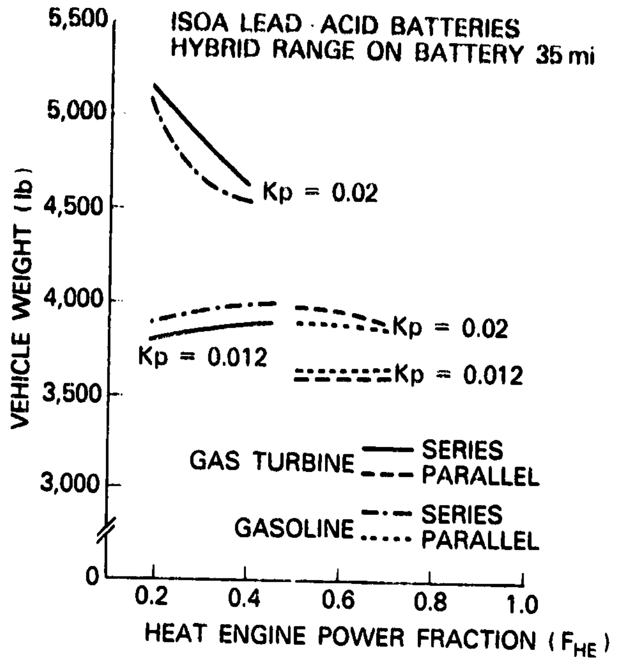


Figure 5-16. Comparison of the Weight of Hybrid Vehicles Using Gasoline and Gas Turbine Engines

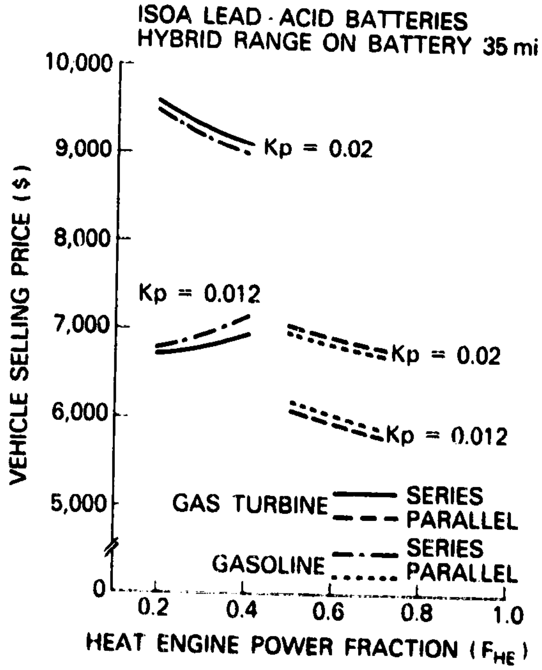


Figure 5-17. Comparison of the Selling Price of Hybrid Vehicles Using Gasoline and Gas Turbine Engines

Figure 5-15 indicates that from the vehicle weight and selling price, the gasoline rotary engine would be attractive for use in the hybrid vehicle. As noted in Section 3.1, there is little doubt that from a packaging viewpoint (bulk and weight) the rotary engine could be used in the hybrid. Unfortunately, the only rotary engine being marketed (by Mazda in the RX-7) is a two-rotor design having 100 hp; it is too large for efficient operation in the hybrid vehicle. A single-rotor rotary engine, if it were available and if questions regarding the fuel consumption and emissions characteristics of the engine were adequately answered, could be used in the present hybrid vehicle design. However, there is no reason to believe that such a single-rotor rotary engine would offer significant advantages compared with a high-speed, four-cylinder European gasoline reciprocating engine.

The two prime engine candidates for use in the hybrid vehicle are the ICE gasoline and diesel engines. Both of these engine types are currently either being marketed in high volume or are in an advanced state of development prior to introduction into the high-volume auto market. In the case of the gasoline engine there are many engines available in the 50-70 hp range for use in the hybrid vehicle. Both carbureted and fuel-injected gasoline engines are available and the emission control systems on these engines are highly developed using both two-way and three-way catalysts. Modification of these engines for the on/off hybrid mode is possible within the time and budget of the NTHV program.

Small, lightweight automotive diesel engines have been developed in the last few years for use in subcompact cars, such as the VW Rabbit. The engines currently marketed by VW are naturally aspirated (NA) but turbocharged (TC) versions of the engine have been developed (4) and are being tested by VW prior to marketing them. Diesel engines are slightly heavier and more expensive (see Figure 5-15) than gasoline engines, but they offer advantages in fuel economy. For this reason they have been considered in detail for the hybrid application. As discussed in Section 8.1, the fuel economy advantage of the diesel was also found in the hybrid vehicle simulations. As is the case for the conventional vehicle, a concern in using the diesel engine in the hybrid vehicle is emissions. Prechamber diesel engines have inherently very low hydrocarbons and carbon monoxide emissions because of their lean combustion. The NO_x emissions from diesel engines are considerably lower than from gasoline engines, but the lean nature of the combustion in the diesel precludes the use of a three-way catalyst to reduce further the diesel NO_x emissions to levels as low as is possible with the gasoline engine (less than 1.0 gm/mi). In addition, diesel engine exhausts contain small (micron-size) carbon particles in sufficient concentration to be of concern. Particulate emissions of 0.5-1 gm/mi are common for most diesel-powered passenger cars. The Environmental Protection Agency is currently considering a particulate emission standard of 0.2 gm/mi for 1983. Current diesel engines are not likely to meet that standard without a trap or other particulate-collection device. The emission calculations discussed in Section 8.1 indicate that in the hybrid application it is likely that the diesel engine can meet an NO_x standard of 1.0 gm/mi but it seems very unlikely that the diesel can meet the proposed 0.2 gm/mi particulate standard even when the battery pack is near full charge. Odor is not expected to be a problem with the diesel in the hybrid application because the engine would not idle or be used at light loads where odor is greatest.

At the present time, both a fuel-injected gasoline and turbocharged diesel engine are being considered in the program. Both engines are available from Volkswagen. These engines are the 1.6-ℓ gasoline fuel-injected engine and the 1.5-ℓ TC diesel engine (currently referred to as a research engine by Volkswagen, but it is in an advanced state of development). Detailed characteristics of both of these engines are given in Section 3.1. Fortunately, both engines use essentially the same block so that there is little or no difference in packaging them in the vehicle. Hence, the packaging layout and even the preliminary design work can proceed without a definite choice made concerning which engine to use. Either engine permits the design of an attractive hybrid vehicle. Most of the vehicle synthesis calculations have been made using the characteristics of the gasoline engine, but both engines are being used in the second-by-second vehicle simulations employing the HYVEC program. Comparisons of the operating cost of gasoline and diesel engine powered hybrid vehicles are made in Reference 31.

5.6 ELECTRIC DRIVE TYPE

There are a number of electric drive systems which can be considered for use in the hybrid/electric vehicle. As discussed in Section 3.2, these drive systems fall into two general categories: (1) dc (direct current) and (2) ac (alternating current). Development work and laboratory testing is currently under way on both dc and ac systems at General Electric Corporate Research and Development and elsewhere. All the electric vehicles presently under development and test in the United States use dc drive systems, but there are proponents of ac drive systems. The advantages and disadvantages of each type of drive system are summarized briefly in the following paragraphs. Much more detailed discussion of electric drive systems is given in Section 3.2 and in Volume II on electric drive systems. In this section the significant trade-offs involved in the choice of the electric drive system are discussed and the effect on hybrid vehicle weight, selling price, and operating cost illustrated using the results of vehicle synthesis calculations.

First consider the dc electric drive system. The major advantage of the dc system is that both the electric motor and battery are direct current devices, and control of the power and speed (rpm) of the motor requires only control of the battery voltage and current as perceived by the motor. All the dc systems considered in this study involved separately excited dc motors in which the armature current and magnetic field current could be controlled separately. Direct current separately excited motors can be operated with both armature and field control or with field control (weakening) only. Using both armature and field control, the torque and speed of the motor can be smoothly controlled down to zero output torque and speed. This electrical drive system behaves as an infinitely variable transmission with infinite speed ratio range. Using only field weakening to control the motor, the motor cannot operate below its base speed (that speed at which the back EMF at maximum field equals the battery voltage). The base speed of the motor can be lowered by reducing the battery voltage. This can be done by rearranging the battery pack connections so that all the batteries are not connected in series. Altering the manner in which the battery pack is connected is referred to as "battery switching." Hence, in the present study of dc electric drive systems, three different systems were considered:

1. Armature and field control

2. Field control without battery switching

3. Field control with battery switching

The major power electronics needed to control a dc motor are associated with the armature chopper which controls the motor voltage. The armature chopper must pass the peak power and current (300-500 A) required by the motor and is by far the most expensive component in the power control system. The field chopper, which controls the field current and thus the magnetic flux in the motor, must pass only a relatively small current (10 to 20 A). Hence, the field chopper is much less expensive than the armature chopper. Field control without battery switching is the lowest cost and most reliable (durable) of the three dc systems, but it requires that the electric motor idle at base speed which is at about 2000 rpm for a separately excited motor having a maximum speed of 6000 rpm and a constant-power speed range or 3 to 1. Wider speed range motors can be built but with significant weight and cost penalties (Reference 34). The simplest form of battery switching, which involves splitting the battery pack into two parallel strings, would permit the voltage to be halved and the motor base speed lowered to about 1000 rpm. This type of battery switching was studied in the present program. The differences in the specific weight and specific costs of the three dc electric drive systems are summarized in Table 5-4.

In simple terms, the primary trade-off involved between dc systems using armature and field control or field control only are ease of vehicle control at low speeds and low torque and cost. A secondary consideration, which will undoubtedly become less of a factor in the future, is system reliability and ruggedness. At the present time, the power transistors used in the armature chopper are in a relatively early stage of development and, as a result, their peak current capability and lifetime are somewhat uncertain. This situation is expected to improve markedly in the years ahead. High-power thyristors could be used in the armature chopper, but this would involve significant penalties in weight and efficiency.

Table 5-4

WEIGHT AND COST CHARACTERISTICS OF VARIOUS DC AND AC ELECTRIC DRIVE SYSTEMS

System	Specific Weight lb/kW	Specific Cost ($/kW) Low Volume*	Specific Cost ($/kW) High Volume†
dc			
Separately Excited Motor	10.00	30.0	20.0
Control			
Field and Armature Control	2.00	21.0	14.0
Field Control	1.25	19.0	12.5
Field Control and Battery Switching	1.50	10.0	6.7
ac			
Induction Motor	7.00	20.0	13.3
Pulse Width Modulated Inverter	3.20	28.5	19.0

*100,000 units/yr

†1,000,000 units/yr

Next consider the ac electric drive system. The major disadvantage of the ac system is that the battery is a dc power source, and the ac motor requires an alternating current and voltage with the frequency dependent on motor rpm. This requires a complex and expensive power conditioning unit (inverter) between the battery and ac motor. The ac inverter is in essence nearly equivalent to three dc armature chopper units in cost and complexity because the ac motor requires three-phase voltage control. The primary attractiveness of the ac electric drive system is that the ac motor can be made smaller, lighter, and lower cost than the dc motor. The size advantage of the ac motor is particularly marked when the possibility of high-speed designs are exploited (see Section 3.2). Unlike the dc motor which can be run without armature control, the ac motor cannot be run without the inverter because the battery produces only dc power. Hence, the low-cost option of an electric drive system without the expensive inverter is not possible with the ac drive system. The specific weight and specific characteristics of the ac system are summarized in Table 5-4.

Vehicle synthesis calculations were made using HYVELD for both dc and ac electric drive systems. The component weight and cost characteristics used in the calculations are given in Table 5-3. The results of the calculations for vehicle weight, selling price, and operating cost are shown in Figures 5-18 and 5-19. From Figure 5-18 it is seen that the effect of electric drive system type on vehicle weight is insignificant (≤25 lbs). In the case of the ac drive, the lower motor weight almost exactly offsets the greater weight of the power conditioning unit. The effect of the different electric drive costs on vehicle selling price and operating cost is shown in Figure 5-19 for vehicles using ISOA lead-acid batteries. Use of the dc drive system without armature control results in a vehicle selling price of about $400 less than projected using more extensive power electronics. The differences in operating cost are about 1¢/mi. It should be noted that the power transistor module cost of $50 used to determine the vehicle selling price results given in Figure 5-19 are the present goals of the transistor program, and whether the goals can be met is not certain. If the power transistor costs are higher, then the price differences between the various electric drive systems would be greater. The effect of the electric drive system type on annual net dollar savings as a function of gas price and volume production rate of the electrical components is shown in Figure 5-20. In using the lowest cost dc electric drive system, a net annual dollar savings is projected for a gas price between 1-$1.5/gal even at the 100,000 units/yr production rate.

Based on the vehicle synthesis results discussed in this section and the second-by-second vehicle simulation results discussed in Section 8, it was decided to proceed into the Preliminary Design Task using the dc electric drive system without armature control. Efforts will be made to design microprocessor control circuits so that battery switching can be implemented with good efficiency and satisfactory vehicle driveability (smoothness) at low vehicle speeds. Careful design of an automatic clutch to initiate vehicle motion at stops is also required if armature electronics is not used.

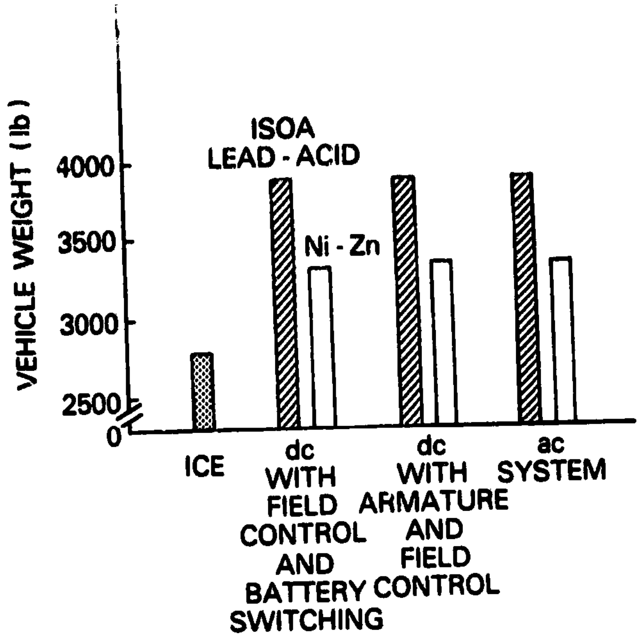


Figure 5-18. Effect of Electric Drive System on Vehicle Weight

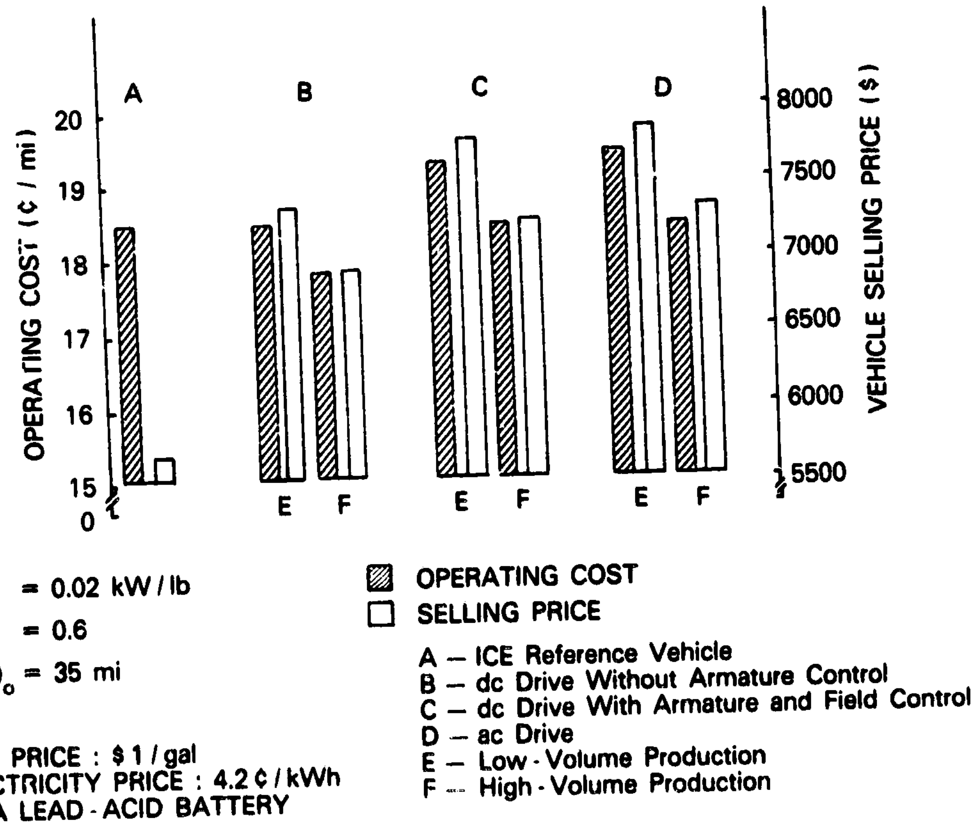


Figure 5-19. Comparison of the Operating Cost and Selling Price of Parallel Hybrid Vehicles Using dc and ac Drives

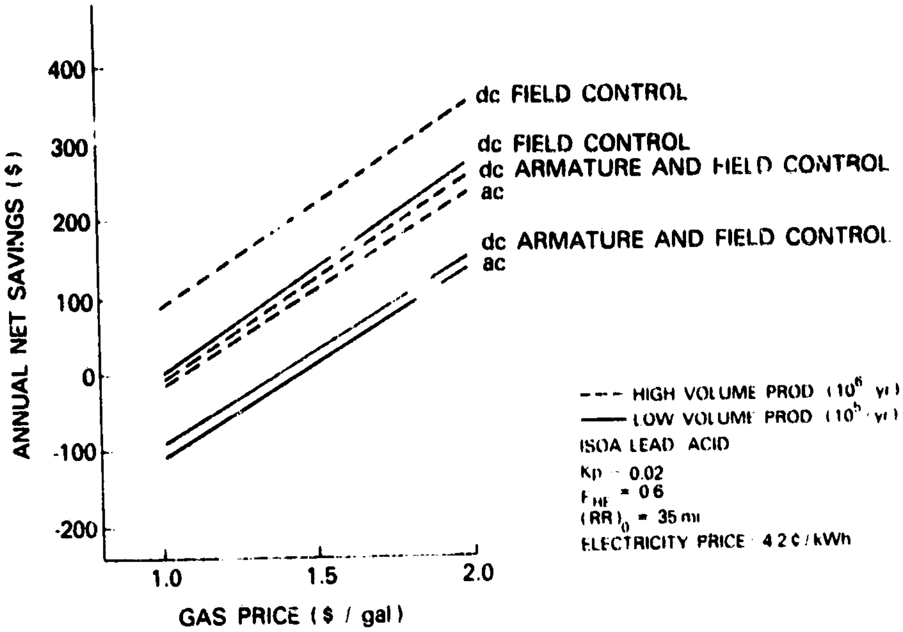


Figure 5-20. Annual Net Dollar Savings as a Function of Gas Price for Various Electric Drive Systems and Volume Production Rates

Section 6

VEHICLE DESIGN ANALYSIS AND LAYOUT TRADE-OFF

Section 6

VEHICLE DESIGN ANALYSIS AND LAYOUT TRADE-OFF

6.1 INTRODUCTION

This section presents the analyses and reasoning used in performing power train packaging and vehicle layout trade-offs. Factors considered and discussed include vehicle weight and drag estimates; accessory requirements, such as for power steering, power brakes, and air-conditioning; and vehicle packaging, such as seating and interior dimensions, as well as engine and battery placement. A configuration rating method was established and used as an aid.

6.2 VEHICLE WEIGHT ESTIMATES

The purpose of this analysis is to provide detailed weight breakdown estimates for the various vehicle arrangements and to determine an appropriate weight propagation factor for use in the design trade-off studies.

Weight breakdowns for thirteen different vehicles were analyzed to establish relationships among vehicle gross weight, number of passengers, engine size, wheel-base, and certain component weights. Vehicle weights were divided into 29 categories for the purpose of the analysis. Weights of each category for the 13 vehicles, where data formats allowed, were studied in order to determine functional relationships between the weights and the independent parameters stated above.

The 13 vehicles studied and their gross weights are listed below:

VEHICLE	GROSS VEHICLE WEIGHT, GVW (lb)
• Dasher Four-door	2923
• Fiat 124	2786
• Fiat 128	2610
• Subaru DL	2410
• Chevette	2781
• Oldsmobile F-83 (1972)	4460
• Chevrolet Corvette	3711
• Chevrolet Vega	3097
• Ford Pinto	3276
• Chevrolet Chevelle (1975)	4443
• Chevrolet Camaro (1974)	4287
• Volkswagen Rabbit	2583
• Buick Regal (1978)	4034

Wc = Vehicle curb wt-lb
Wg = Vehicle gross wt-lb
p = Number of vehicle passengers
L = Vehicle wheelbase - in.
F = Fuel system capacity - gal
D = Engine displacement - in.^{3}

6.2.1 DERIVED RELATIONSHIPS

The following relationships were developed using the least squares technique for the various vehicle elements.

1. Gross vehicle weight related items

Item	Relationship
Structural	0.178 Wg
Bumpers	0.00147 Wg + 0.0000101 Wg^2
Suspension	0.02526 Wg + 0.00000736 Wg^2
Wheels & tires	20 + 0.04666 Wg
Brakes	9.2 + 0.02368 Wg
Tools	0.00281 Wg
Total	29.2 + 0.2779 Wg + 1.74 x 10^{-5} Wg^2

2. Engine displacement related items

Engine and transmission	290 + 0.91D
Exhaust system	9 + 0.19D
Cooling system	24 + 0.08D
Total	323 + 1.18D

3. Passenger-related items

Seats and related	23.2P

4. Wheelbase-related items

Skins	1.4L

5. Fuel-capacity-related items

Fuel system (including evaporative emission control)	1.53F

6. Insensitive components

Certain items are relatively insensitive to any of the variables considered in the context of this study. These items deal primarily with human factors (i.e., elements designed to tolerate loads supplied by the driver). Those items which will be considered fixed in weight are listed below:

Item	Weight (lb)
Doors (four-door) and deck	174.0
Exterior trim	4.5
Instrument panel	20.0
Interior trim	60.0
Windshield wipers	7.6
Fixed glass	48.0
Park brake	5.1
Brake actuation	5.3
Brake hydraulics	15.8
Controls	9.8
Power steering gear	30.6
Steering linkage	19.8
Steering column and wheel	16.6
Hydraulics	14.7
Accelerator electrical	59.5
Heater	15.5
Restraint system	31.8
Air conditioner	126.0
Total Weight	664.6

6.2.2 PROJECTIONS OF FUTURE PRODUCTION VEHICLE WEIGHTS

As a result of conversations with knowledgeable people within the automobile industry, the consensus of opinion was that a reduction in vehicle component weights by 1985 will not exceed 10 to 12% compared to the same sized 1979 vehicle. This weight reduction will come from basic redesign of components with the use of more specialized parts (less commonality across vehicle lines) rather than by the substitution of other materials for specific parts. Disadvantages to shifting to aluminum or plastic components in lieu of steel are:

- Cost effectiveness ($1.00 or more penalty per pound)
- Requirement for significant investment in tooling and equipment
- Inadequate supply of aluminum or resin products to meet potential automotive requirements

6.2.3 WEIGHT PROPAGATION FACTORS

Vehicle weight propagation factors for the purpose of performance trade-offs can then be derived from this analysis. Differentiating the relationship for gross vehicle-weight-related elements, and evaluating that slope at a nominal gross weight of 4500 lb, the weight propagation factor (rate of change in vehicle weight relative to the change in gross weight rating) is 0.42. It should be noted that this factor includes additional weight added to other chassis items in the vehicle. If the future reduction in component weights as outlined in Section 6.2.2 are applied to the appropriate items of the gross vehicle-related elements, the weight propagation factor is reduced to 0.375. Future reductions in weight were not applied to the tires and to the brakes. Thermal considerations, rather than stress limits, determine the size and weight of the brakes.

6.2.4 PROJECTED HYBRID VEHICLE WEIGHT

Preliminary weight projections for the Hybrid Vehicle were made with the following assumptions:

- Chevrolet Malibu seating package
- Lead-acid batteries (700 lb)
- Twelve-gallon fuel tank
- Volkswagen 1.6-ℓ engine
- Four-speed automatic transmission
- General Electric/Chrysler near-term electric motor and control (360 lb)

Table 6-1 itemizes this analysis. Weights for the 1979 hybrid were based on the relationships generated for the weight propagation factor study. Weights for the 1985 hybrid were derived from weight reductions gained through the substitution of materials and the redesign of elements as discussed in Section 6.2.2.

6.3 VEHICLE DRAG PROJECTIONS

6.3.1 INTRODUCTION

For the purposes of the trade-off analysis, a drag estimate must be made for the hybrid vehicle. The basic elements of the total vehicle drag which will be considered are the aerodynamic drag, tire hysteresis losses, the tire rolling resistance, and chassis losses.

6.3.2 AERODYNAMIC DRAG

Vehicle frontal area and drag coefficient must both be established in order to determine the total aerodynamic drag. The frontal area is primarily a function of the seating package selected. With the 1979 Chevrolet Malibu seating arrangement as the model, a frontal area of 21 ft^2 is a reasonable estimate.

Table 6-1

WEIGHT ANALYSIS - MALIBU BASED HYBRID

Item	Weight (lb) Malibu (estimated)	Projected Variation	1979 Hybrid	1985 Hybrid
Structural	718	+ 178	896	806
Bumpers	170	+ 13	183	164
Suspension	251	+ 6	257	230
Wheels and Tires	208	+ 46	254	254
Brakes	105	+ 23	128	128
Tools	11	+ 3	14	14
Subtotal	1463	+ 269	1732	1596
Engine and transmission	552	- 278	274	274
Exhaust	47	- 20	27	27
Cooling	40	- 9	31	31
Subtotal	639	- 307	332	332
Seats and related	116	0	116	104
Skins	170	0	170	153
Fuel system	27	- 9	18	18
Insensitive	538	0	538	484
Air conditioner	126	0	126	113
Subtotal	977	- 9	968	872
Batteries	0	+ 700	700	700
PCU	0	+ 360	360	360
Subtotal	0	+1060	1060	1060
Curb weight	3079		4092	3860

As demonstrated by the full-scale wind tunnel tests of the General Electric Centennial-100 vehicle, a drag coefficient of about 0.33 is possible. This value should be adjusted by two factors. First, the fact that the hybrid vehicle will require a radiator for its internal combustion engine will increase the drag by approximately 6% to 0.35. Secondly, the aerodynamic performance of the vehicle in yaw must be considered since the vehicle will seldom operate in a zero wind condition. Making the assumption that a 10 mph wind is typical and the average speed of the vehicle is about 40 mph, a yaw angle range of +15° is reasonable. During scale wind tunnel tests of the General Electric Centennial in

December of 1975, the effect of the yaw angle on the drag coefficient was determined to increase it by 11%. Hence, a reasonable estimate of the drag coefficient for the hybrid/electric vehicle is 0.39.

6.3.3 TIRE ROLLING LOSSES

Tire rolling resistance for radial-ply tires in equilibrium operating conditions are 0.011 lb per lb at their rated load. However, the effect of warmup is significant and can account for variations of up to 50% in tire rolling loss from "cold" to "warm" conditions. This factor should be considered with the value selected dependent on the mission driving profile. Based on data from the report "Tire Rolling Loss Measurements" by Calspan Corporation under contract to the Department of Transportation, the influence of vehicle speed on tire losses was determined to be

$$F = 0.00003\ WV$$

where F = Drag force - lb
W = Weight - lb
V = Speed - mph

6.3.4 TOTAL VEHICLE DRAG (AERODYNAMIC AND TIRES)

Summing the effects outlined above, the expressions for the drag of the hybrid vehicle in the equilibrium condition (warm tires) can be written:

$$\begin{aligned} F &= 0.011\ W + 0.00003\ WV + (0.00249)\ (A_f C_D)\ V^2 \\ &= 0.011\ W + 0.00003\ WV + 0.02\ V^2 \end{aligned}$$

where W = Vehicle weight - lb
V = Vehicle speed - mph
A_f = Vehicle frontal area - sq ft
C_D = Drag coefficient

6.4 VEHICLE ACCESSORIES

6.4.1 INTRODUCTION

As stated in the original proposal, the reference ICE vehicle will be used to establish the performance requirements for the accessory systems in the hybrid/electric vehicle. In order to determine these values, a 1979 Chevrolet Malibu four-door sedan was tested. All the data obtained is included in Volume II of this report. The results of the tests are summarized and their impact on the hybrid vehicle design discussed in the following sections.

6.4.2 ACCESSORY SYSTEMS

The accessory systems which will be considered are listed below, along with the method of providing for these functions in the Reference Vehicle.

System	Power Source
Power steering	Hydraulic (open centered)
Power brakes	Engine manifold vacuum
Air conditioner	Engine-driven compressor
Lighting	Engine-driven alternator
Heater and defroster	Waste engine heat and alternator
Windshield wiping	Engine-driven alternator
Transmission clutching (automatic)	Hydraulic (open centered)
Comfort and convenience	Engine-driven alternator

Each of these systems must be included in the hybrid vehicle if it is to have the same value to the customer as the Reference Vehicle.

6.4.2.1 Power Steering

The power steering system on the 1979 Malibu is an open-centered hydraulic system. The horsepower required to drive the pump was measured using a strain-gauged crankshaft pulley to measure drive torque and an electronic tachometer to measure engine speed. Figure 6-1 is a plot of power steering pump horsepower requirements as a function of pump speed. Maximum requirements of the system are in the parking mode which occurs at approximately 1200 pump rpm. Maximum flow of the system is three gallons per minute. Power required in this condition is 3.25 hp. Tests run by Saginaw Steering Gear Division of General Motors indicate that the average power consumption of the power steering system on the city driving cycle is approximately 0.75 hp.

Two basic systems for power steering were considered: one is the conventional open-centered hydraulic system found on the Chevrolet Malibu, while the other is a closed-center system currently used on the French Citroen. The primary difference between these two systems is that the closed-center system consumes little energy when it is not in use. Oil is stored under pressure in an accumulator and used to supply the peak demands of the system (see Figure 6-2). Noting that the average speed in the Environmental Protection Agency (EPA) urban cycle is approximately 20 mph and that the no-load pump power is approximately 0.35 hp at that speed, the average power required to pump the hydraulic fluid is about 0.40 hp. This would be the average power required by the closed-center system during the urban cycle.

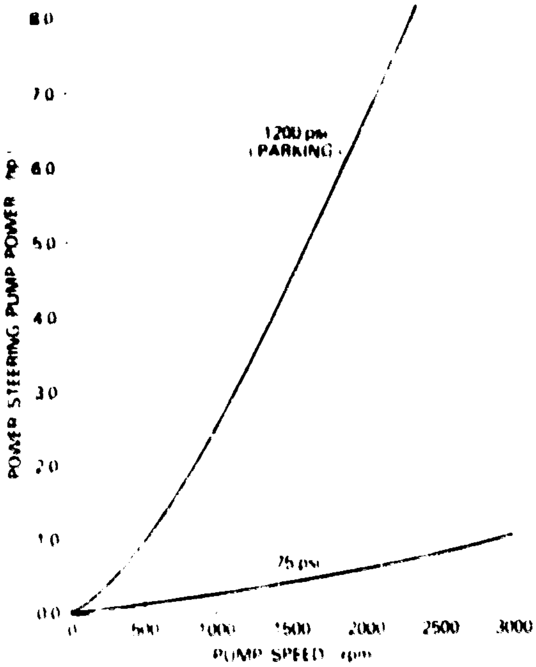
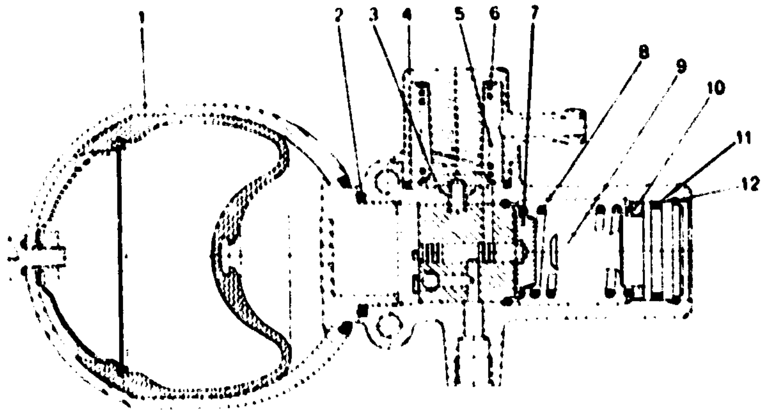


Figure 6-1. Power Steering Pump Power Requirements, 1979 Malibu



1	Main accumulator	7	Spring cup
2	Ring seal	8	Cut-in spring
3	Spring flange	9	Spring plug
4	Spring housing	10	Spring adjusting shims
5	Cut out spring	11	Ring seal
6	Spring adjusting shims	12	Snap ring

Figure 6-2. Main Accumulator and Pressure Regulator

Due primarily to the large energy savings and the load-levelling effect of the accumulator which reduces the maximum pump power requirements, the closed-center system is favored for the hybrid vehicle application.

6.4.2.2 Power Brakes

Brake system boosting in most contemporary automobiles is provided by the use of engine inlet manifold vacuum. Since the ICE is run intermittently and additionally run at higher loads (lower manifold vacuum) and never in over-run (high manifold vacuum) situations, this method is not considered applicable for the hybrid vehicle.

Alternate systems that were considered are open-centered hydraulic booster (currently available on Lincoln Continentals) or closed-center hydraulic systems utilizing a metering valve similar to that used by Citreon for their direct apply system. Of the two, the closed-center system is preferred because the stored energy feature will provide additional backup in the event of a partial system failure.

6.4.2.3 Air Conditioner

To match the "pull down" performance offered by the Chevrolet Malibu and to meet the steady-state passenger compartment temperatures of 70 °F with 100 °F outside air temperature, a heat transfer of approximately 15,000 Btu/hr is required (see Figure 6-3). Measurements in a 1979 Malibu indicate that 3 to 4 kW is required to deliver that performance. When less cooling is required, the two methods customarily used are to cycle the compressor on and off or to throttle the compressor. The on-off control is more efficient and is used in most American cars. An additional possibility is to control the speed of the compressor, but that approach does not seem attractive because the coefficient of performance of the air conditioner decreases significantly at the lower speeds.

The energy required to provide air-conditioning for the hybrid/electric vehicle can be estimated from Figure 6-3. The data given in Figure 6-3 indicate that after the car interior heats up to about 120 to 140 °F it takes about 10 to 15 min for the air conditioner working at maximum capacity to reduce the passenger compartment temperature to 80 °F. Using a cooling rate of 250 Btu/min and an effective coefficient of performance of 2.0 for the air-conditioning system, this requires an input power of 2.2 kW. The energy used in 15 min is then about 0.5 kWh. Figure 6-3 also indicates that for an ambient temperature of 100 °F and a vehicle speed of 40 mph, the maximum air conditioner output is required to maintain the passenger compartment temperature at 75 to 80 °F. Hence, on a very hot and sunny day, the air conditioner would use at least 2.2 kWh/hr of travel. As discussed in Reference 35, the cooling load decreases rapidly as the ambient temperature and average insolation is reduced. In that case, the air conditioner would cycle on and off and the energy use per hour of travel would be much lower than 2.2 kWh/hr.

The energy use of the air-conditioning system is likely to be considerably greater than the minimum values indicated in the

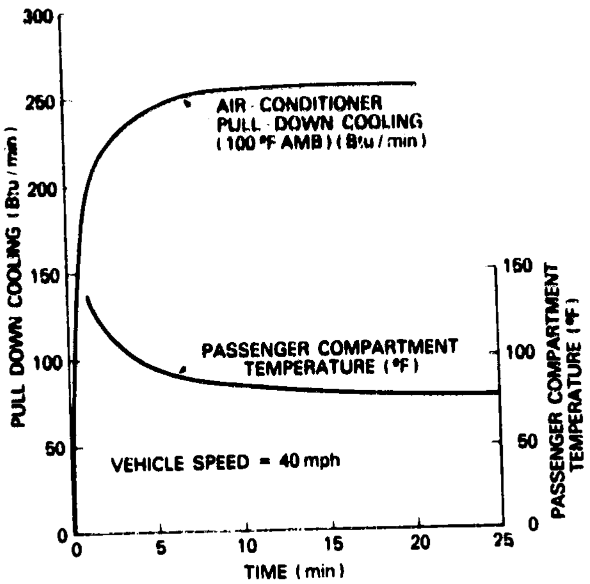


Figure 6-3. Air Conditioner Performance in the 1979 Chevrolet Malibu

previous paragraph because the power required to drive the compressor increases as its shaft rpm increases (Figure 6-4). Unfortunately, the coefficient of performance of the air-conditioning system decreases as the shaft rpm increases and the net effect is only a slight increase in cooling capacity at the higher compressor shaft speeds. Hence, most of the additional power required by the compressor is dissipated in heat and represents a loss. The speed range over which the compressor operates can be greatly reduced by using a variable speed-ratio drive to connect the compressor with the motor output shaft. Such a drive system has been developed by the Morse Chain, Division of Borg-Warner for use in conventional ICE vehicles. The Morse accessory drive maintains a constant output speed for a 2:1 range of input speeds and has a 1:1.45 speed ratio for higher input speeds (see Figure 6-5 taken from Reference 36). A mini-CVT with a speed ratio range of 3:1 or 4:1 could probably be developed using the steel-belt approach being used by Borg-Warner for the automotive transmission. In using the constant-speed drive for the air conditioner, the average power required would be 3.0 kW or 3 kWh/hr of travel time with maximum cooling. For conditions in which the air conditioner was cycling on and off, the energy usage would be proportionately less.

It appears from the above discussion of air conditioner energy usage that it is possible in some circumstances to operate the air conditioner in the hybrid/electric vehicle from the electric motor shaft and thus use battery-stored electricity. In those cases in which maximum cooling is needed for extended periods of time (i.e., no cycling of the compressor), it seems advisable to operate the air-conditioning system off the engine even if that

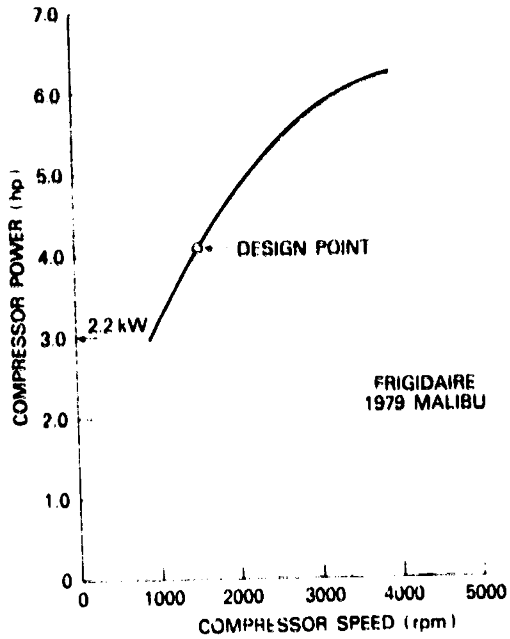


Figure 6-4. Air-Conditioning Compressor Power Requirements

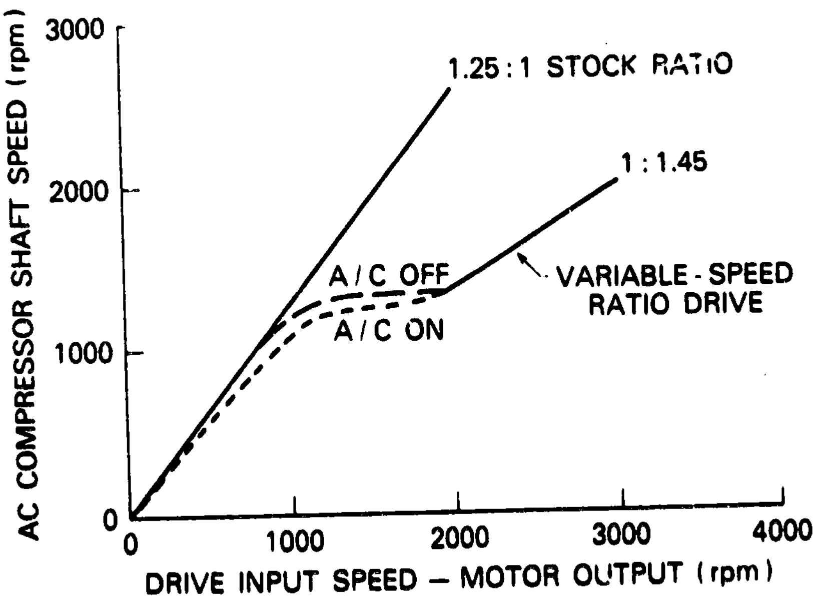


Figure 6-5. Morse Chain Controller Speed Accessory Drive

means idling the engine for that special purpose. Otherwise, the electric range of the hybrid would be significantly reduced making it necessary to charge the batteries from the heat engine. Optimization of engine use to help provide air-conditioning clearly requires further study.

6.4.2.4 Heating and Defrosting

To maintain similar heating performance to the 1979 Malibu, a maximum heat flux of approximately 60,000 Btu/hr is required (see Figure 6-6). This heat flux is required for an ambient temperature of -20 °F. With the assumption that the heat requirement is directly proportional to outside air temperature, the corresponding heat flux requirements are 715 Btu/min for 0 °F and 250 Btu/min for 32 °F. Normally, these heat flux requirements are easily provided by the waste heat from the ICE. In the hybrid/electric vehicle, with the heat engine running less than 30% of the time when the batteries are well-charged, there is not enough waste heat available to meet heating and defrosting requirements. For example, if the hybrid/electric vehicle achieved 70 mpg on the EPA urban cycle (1372 s in duration), the waste heat would only be 4020 Btu and about 6000 Btu would be required at 32 °F. Past experience has also indicated that there is insufficient waste heat from cooling the electric motor and controller to utilize in passenger compartment heating.

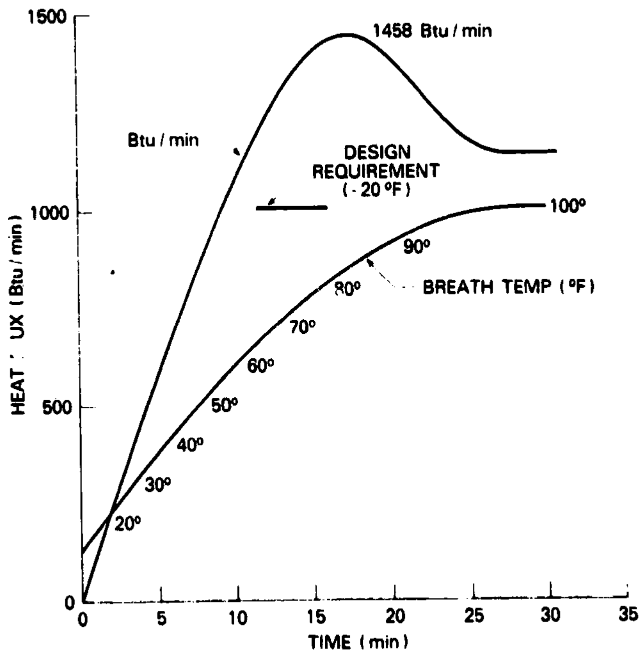


Figure 6-6. Heating Requirements, 1979 Malibu

A suitable method for providing additional heat would be to augment the waste heat available from the ICE with a gasoline burner. For the case of a hybrid/electric which gets 70 mpg on the EPA urban cycle and an ambient temperature of 0 °F, the heat engine would provide 4000 Btu and the gasoline burner 12,000 Btu to heat the passenger compartment. The gasoline required by the gasoline heater would be about 0.10 gal/EPA cycle. This would reduce the vehicle fuel economy on the EPA cycle from 70 to 37 mpg. This is indeed a large reduction in fuel economy. However, it is less than would have been experienced if the engine had been run more often to generate waste heat. The 1.6-ℓ VW gasoline engine has an idle fuel flow of 0.4 gal/hr and would use 0.15 gal/EPA cycle and generate approximately only 9500 Btu in waste heat. The gasoline heater is by far the best means of generating heat for heating and defrosting. It is interesting to note that 715 Btu/min converts to 12.5 kW so that using stored electricity is clearly not feasible.

6.4.2.5 Electrical Accessories

The accessory electrical loads were measured on a 1979 Malibu and are presented in Table 6-2. System operating voltage is 14.6 V.

Table 6-2

ACCESSORY ELECTRICAL LOADS

Accessory	Power Requirement (watts)
Parking lights	101
Low-beam headlights	203
High-beam headlights	254
Turn signals (average)	84
Hazard lights (average)	179
Interior lights	45
Windshield wipers	
dry - low speed	98
wet - low speed	90
dry - high speed	83
wet - high speed	70
Ventilation fan	
Low speed	32
Second speed	73
Third speed	112
High speed	159
Rear window defogger	231
Radio	10
Cigarette lighter	62
Horn	25
Engine ignition system	25
Air conditioner clutch	44

Maximum accessory loads may be established by summing the power requirements for items which can be utilized simultaneously. These include:

Item	Load (watts)
High-beam headlights	254
Windshield wipers	90
Ventilation fan	159
Rear window defogger	231
Engine ignition	25
Radio	10
Reserve capacity	75
Maximum Accessory Load	844

At 14.6 V, which is the regulated operating voltage in the Malibu tested, the current rating of the system should be a minimum of 58 A.

Three methods were considered to provide this accessory power. First, the devices could be redesigned to operate at the power system voltage. This was not determined to be practical due to the durability requirements of the lamp filaments. Higher system voltages would result in a much more fragile lamp. The second method would employ a direct current to direct current converter making use of the main battery as its energy source. This device would be required to provide up to 58 A continuously at 14.6 to 15.8 V with input voltage swings from 140 to 80 V to account for both regeneration and acceleration voltages.

This direct current to direct current converter would need higher capacity than the one recently developed for the GE/DOE Near-Term Electric Car and would require further development. The third potential method is the conventional alternator. Devices are currently in production and are capable of providing this level of power. Figure 6-7 is a plot of alternator power requirement as a function of speed and current. Operating voltage is 14.6. The ratio of alternator speed to car speed is 111.6 rpm/mph.

If it is assumed that a nominal accessory load is 25 A (a reasonable combination of accessories), the average power consumed throughout the urban driving cycle is approximately 1.0 hp. The efficiency of the charging system at this point of operation is approximately 49%. It may be possible to improve this efficiency by a slight redesign or sizing of the alternator.

The final choice of an approach to meet electrical accessory loads is to further investigate the cost, efficiency, and practicality of a direct current to direct current converter with the required rating. Since this would be a new development, the alternator is considered the prime candidate in the trade-off studies.

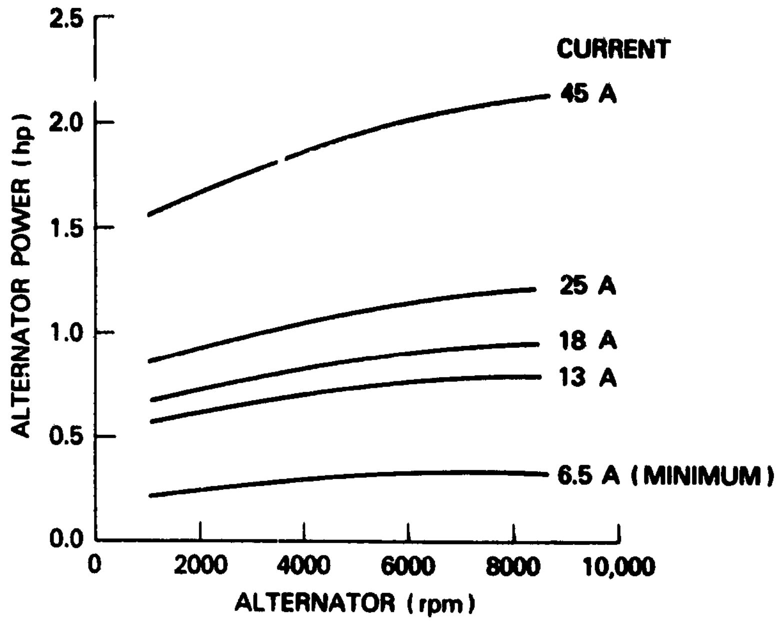


Figure 6-7. Alternator Power Requirements, Delcotron - 1979 Malibu

6.4.2.6 Transmission Control

Transmission control is a function that is often not considered a vehicle accessory, but if torque converters are not employed, some method of providing hydraulic pressure for transmission shifting independent of the transmission input shaft must be included. Present automatically shifted gearboxes utilize an open-centered hydraulic system to provide this energy. In addition to the shifting requirements, there are also lubrication and cooling functions to be included. To minimize hydraulic pumping losses, closed-center operation of the transmission shifting function will be considered. Since this modification will require some development activity, additional investigation in the preliminary design phase will be required. The potential power savings in the change from open to closed-center shifting control is in the order of 0.5 kW.

6.4.3 ACCESSORY DRIVE OPTIONS

Accessory loads in conventional ICE vehicles are driven off the engine. This is functionally practical since the ICE is running during all modes of vehicle operation. The parallel hybrid configuration poses some unique problems since power may be delivered to the vehicle from either or both of the primary power sources. Additionally, some parallel systems could result in neither the heat engine nor the electric drive motor running when the vehicle is at rest. Unfortunately, there are requirements for accessory power when the vehicle is at rest. These are:

- Power steering for parking
- Power brakes for holding on grades
- Air-conditioning for pull-down and for traffic jam situations
- Lighting (to maintain minimum voltage)
- Heater/defroster

There are several available options to provide the mechanical power required. These are:

- Idle the heat engine
- Provide a separate electric motor for accessories
- Store energy in the proper form for use when the vehicle is at rest
- Idle the electric motor

Some system to augment the 12-V accessory battery is required at idle in order to maintain the system voltage at its allowable minimum (13.8 V) for lighting. The total maximum continuous requirement consists of the following:

Element	Horsepower
Air-Conditioning	4.0
Alternator	1.0
Power steering	2.4
Total	7.4

6.4.3.1 Analysis of System Alternates

Engine Idling. Vehicle simulation calculations indicate that for the hybrid/electric vehicle the increase in fuel comsumption caused by the engine idling is in the order of 50% when the vehicle is operated primarily on electricity and, therefore, is not considered a reasonable option.

Separate Electric Motor for Accessories. The advantages of the use of a separate electrical machine to drive accessories are the ease of speed control and the ability to be independent of the complex hybrid drive system. The use of a separate electric motor also has the potential advantage of utilizing more electrical energy than gasoline during highway operation.

The disadvantages of this system are, of course, weight and cost. The approximate weight of a shunt-wound dc motor rated at 5.5 kW and its associated starting devices is 95 lb, and the cost is in the neighborhood of $150.00. Clearly, this is a large penalty to pay.

Energy Storage Devices. Storing energy for accessories would work satisfactorily in "driving cycle" situations where the idle periods are brief. In the real world, however, when prolonged idling is possible, e.g., traffic jams, it is not practical, to store enough energy in a hydraulic accumulator to provide all of the required steering and brake functions.

Idling the Electric Motor. Idling the electric motor appears to be the best solution to the accessory drive problem. As a large machine running at its base speed, it will consume relatively low power, and, in fact, will operate more efficiently than a smaller separate accessory motor operating at its rated power. Additionally, starting and control devices which would be duplicated in the separate motor case, would be shared with the vehicle starting system.

6.4.4 SUMMARY

Figure 6-8 is a block diagram representation of the accessory systems as presently envisioned.

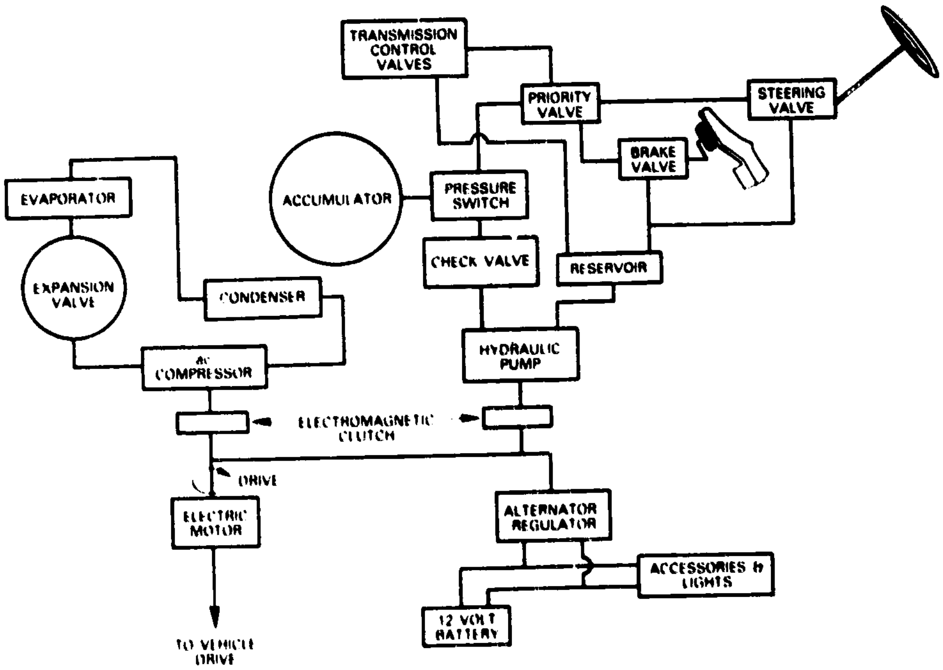


Figure 6-8. Block Diagram of Accessory Systems

The hybrid/electric vehicle will be capable of providing all of the accessories normally found on the reference ICE vehicle. Mechanical power to drive the necessary devices will be delivered from the electric drive motor which will idle during the time that the vehicle is at rest.

Power steering, power brakes, and transmission shifting functions will be provided by a closed-center hydraulic system with the pump driven on demand from the electric-drive motor. Air-conditioning will be of the vapor-compression type with cycling clutch control driven from the electric drive motor and/or the heat engine.

Accessory electrical system power will be delivered from a conventional alternator driven from the electrical drive motor with a fixed ratio. A small accessory battery will be placed across the alternator circuit to provide field excitation, to improve regulation and to provide emergency power in the event of a failure.

The heating system will utilize a conventional hot water system augmented by a gasoline burner for periods of extensive use of the electric drive system and for prolonged idle time.

6.5 VEHICLE PACKAGING

6.5.1 INTRODUCTION

Packaging studies were conducted utilizing the 1979 Chevrolet Malibu as a reference. For purposes of these studies, the interior package was left intact. Additionally, other elements such as the ground clearance, ramp and departure angles, and break-over considerations were maintained. The following sections define the starting points, establish the major packaging considerations, and discuss some of the arrangements evaluated.

6.5.2 PACKAGING CONSIDERATIONS

Based on the design trade-off studies and on the requirements for equal consumer value, the hybrid vehicle packaging studies proceeded with the following basic guidelines:

- Seating package - 1979 Malibu four-door sedan
- Engine - four-cycle gasoline or diesel (70-80 bhp)
- Batteries - lead-acid (700 lbs)
- Electric motor - GE 2364 (20 kW continuous rating)
- Transmission - four-speed automatically shifted or CVT
- Fuel capacity - 12 gal
- Drive arrangement - parallel hybrid with differential prime mover input
- Emission control - three-way catalyst (gasoline engine)
- Accessories - air conditioner, closed-center hydraulic system, alternator driven from electric motor
- Luggage compartment volume - 17 ft^3

6.5.2.1 Seating and Interior Dimensions (Figures 6-9 and 6-10)

According to the basic plan outlined in the original proposal, the interior dimensions of the Reference ICE Vehicle as relating

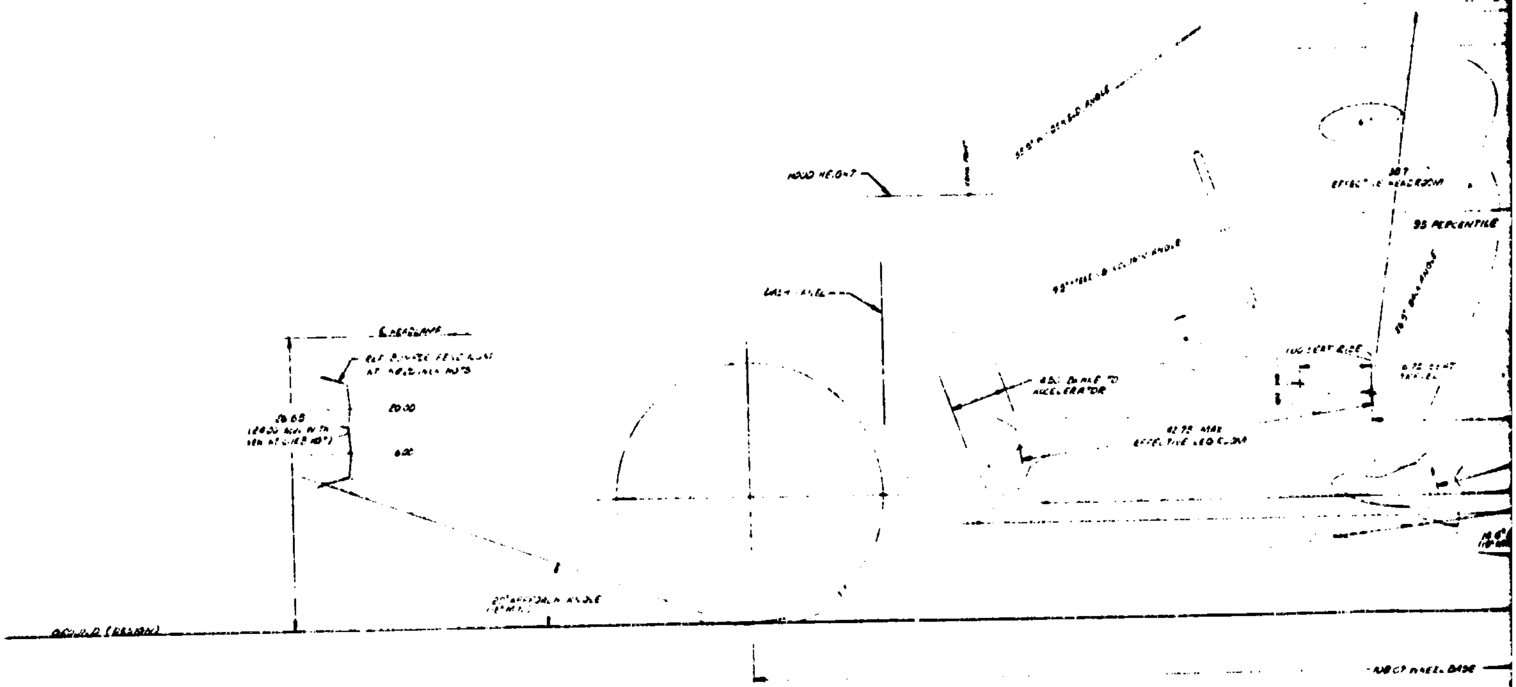
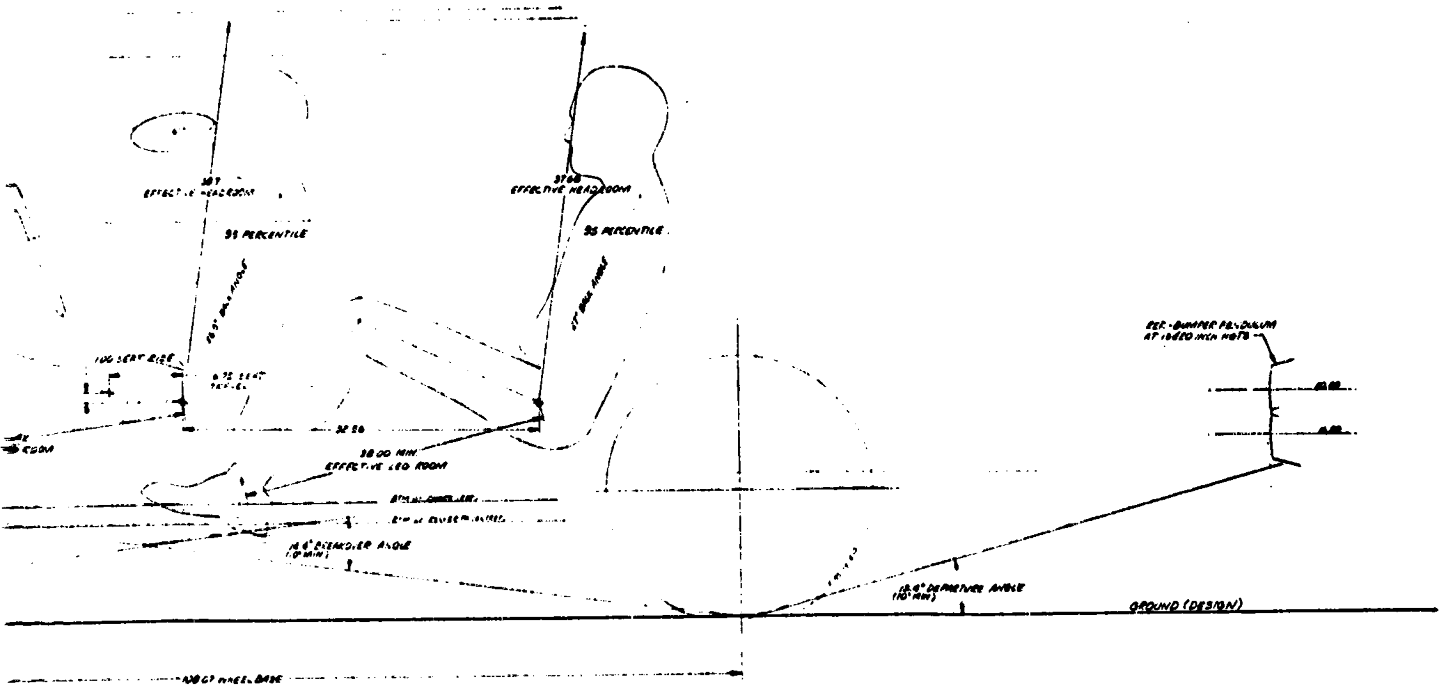


Figure 6-9. Preliminary Layout



eliminary Layout-Space Relation

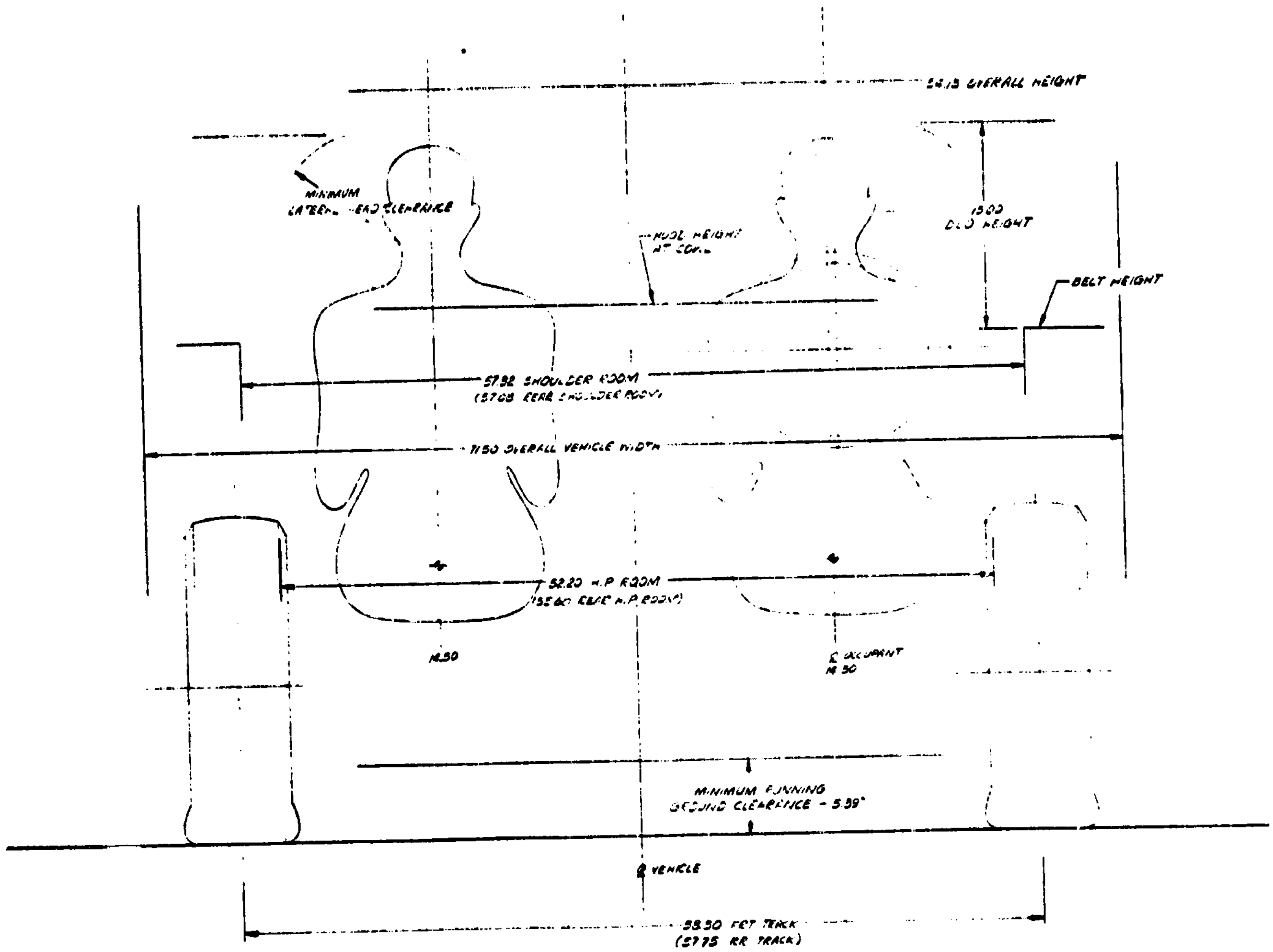


Figure 6-10. Preliminary Layout-Space Relation

to the occupant-seating package would be utilized in the hybrid vehicle. The interior dimensions of the reference ICE vehicle (1979 Malibu four-door sedan) which will be used in the preliminary packaging studies are listed below:

Front Compartment

		Degrees	Inch	mm
W20	Centerline occupant to centerline car		14.48	368
H61	Effective headroom		38.70	983
L64	Maximum effective legroom		42.75	1086
H30	H point to heel hard (chair height)		8.97	228
L40	Back angle	26.5		
L42	Hip angle	99.5		
L44	Knee angle	131.0		
L46	Foct angle	87.0		
L53	H point to heel point		35.07	891
L17	H point travel		6.73	171
H58	H point rise		0.98	25
W3	Shoulderroom		51.32	1456
W5	Hiproom		52.20	1326
W16	Seat width		49.49	1257

Rear Compartment

		Degrees	Inch	mm
L50	H point couple		32.56	827
W25	Centerline occupant to centerline car		13.27	337
H63	Effective headroom		37.68	957
L51	Maximum effective legroom		38.00	965
H31	H point to heel point (chair height)		11.73	298
L41	Back angle	27.0		
L43	Hip angle	92.0		
L45	Knee angle	102.0		
L47	Foot angle	118.5		
W4	Shoulderroom		57.08	1450
W6	Hiproom		55.59	1412

Control Location

		Degrees	Inch	mm
H18	Steering wheel angle	19.5		
L7	Steering wheel torso clearance		13.38	340
L13	Brake pedal knee clear		24.42	595
L52	Brake pedal to accelerator		4.48	114

6.5.2.2 Engine

The engine selected for use in the preliminary packaging studies is the 1.6-ℓ Volkswagen four-cylinder gasoline engine. This engine develops a maximum of 78 hp at 5800 rpm. The VW turbocharged diesel utilizes the same engine block and hence has nearly the same exterior profile as the gasoline engine.

6.5.2.3 Batteries

Lead-acid batteries were used in the packaging studies. No particular case dimensions were used for this study. It was felt that at the proposed manufacturing volume, any reasonable case shape would be manufactured. Battery density was assumed to be 0.077 lb per $in.^3$, including space for wiring, caps, and ventilation provisions.

If Ni-Zn batteries should become available in time for the Near-Term Hybrid Vehicle Program, they can easily be accommodated because they will be somewhat smaller and lighter. The present design is being made to handle the worst case.

6.5.2.4 Electric Motor

The electric motor and power conditioning components were essentially those utilized in the current General Electric/Department of Energy Near-Term Electric Vehicle Program. If battery switching is used rather than armature control, the volume and weight of the power electronics will be less. Packaging space is allotted for the worst case.

6.5.2.5 Transmissions

Two different transmissions were utilized in the preliminary packaging studies. These are four-speed automatics manufactured by Borg-Warner for use in European vehicles. One of them is a conventional drive-through type, while the other is a through-and-back arrangement employed in transverse arrangements. Space is also available for use of the Borg-Warner Transmatic Steel-Belt Continuously Variable Transmission.

6.5.2.6 Drive Arrangement

The drive arrangement selected was of the parallel type. A schematic representation of the basic drive arrangement is illustrated in Figure 6-11. For the purposes of the preliminary packaging studies, a power blending differential gearset and the associated torque reaction clutches have been included. More detailed analysis will be included in the preliminary design phase which will establish the value of this element as compared with the simpler single-shaft drive arrangement.

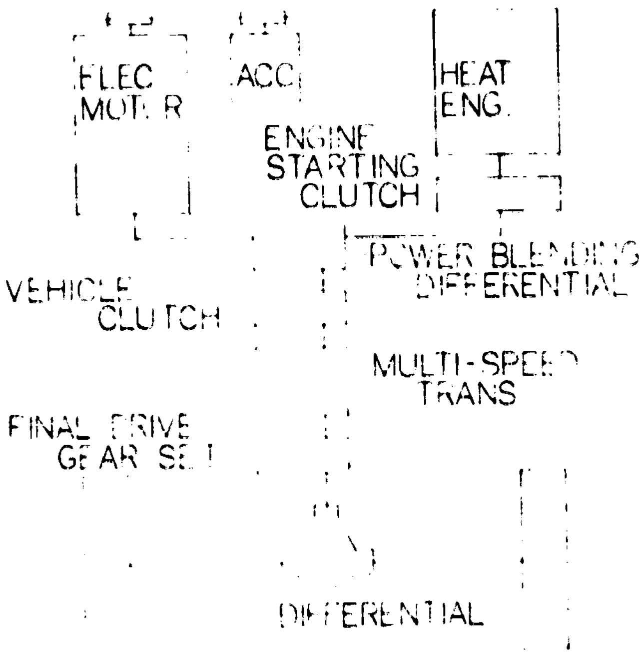


Figure 6-11. Schematic Drive Arrangement

6.5.3 CRITERIA FOR PACKAGE SELECTION

6.5.3.1 Functional Considerations

The primary, and certainly the most basic consideration, is that of functionality. If it doesn't function properly, other considerations like safety, legality, or cost will not influence consumer acceptance. All of the arrangements considered herein were judged functionally adequate.

6.5.3.2 Passenger Compartment Intrusion

Arrangements were judged more or less acceptable depending upon the amount of physical intrusion into the occupant compartment for elements like bumps and tunnels. No interference with any of the seating positions was allowed.

6.5.3.3 Compliance with Federal Standards

Consideration was given to all Federal motor vehicle safety standards in areas that reflect on the basic package. Examples of this compliance are elements like bumper position and clearance, lighting, selection of wheels and tires, and provision for required emission control devices.

Certain of the Federal standards have more profound implications and will be discussed separately.

6.5.3.4 Occupant Crash Protection (FMVSS 208)

The stated purpose of this law is to "reduce the number of deaths of vehicle occupants and the severity of injuries, by specifying the crashworthiness requirements" of the vehicle. There are really two implications of this law. The first essentially defines the requirements of the interior of the vehicle and occupant restraint system. For the purposes of this project, the reference ICE vehicle restraint system and interior compartment will be considered adequate under FMVSS 208. No occupant kinematics investigations will be carried out under this contract since they have no bearing on the hybrid nature of the vehicle drive-line.

A further implication of FMVSS 208 is that the vehicle structure be adequate to provide the integrity to the passenger compartment implicitly demanded by the standard. This is the golden rule of packaging for crashworthiness - maintain the integrity of the passenger compartment. This is accomplished by providing sufficient space around "hard items" to keep them out of the passenger compartment when the structure of the vehicle is dynamically crushed, and by restraining components which are behind the passenger compartment with sufficient structure to prevent impingement on the passenger compartment.

The vehicle structure must also provide a good "ride down" which means that adequate crush distance must be provided in conjunction with structural strength designed to provide deceleration levels commensurate with the restraint system employed.

6.5.3.5 Fuel System Integrity (FMVSS 301)

Fuel containment requirements of this standard state that no more than five ounces of fuel by weight shall spill during the five minute period following a barrier crash, a rear moving barrier, or a roll-over. The packaging implications of such a standard are clear. Locate the fuel in such a position that the tank will not be severely damaged in collision situations.

6.5.3.6 Battery Crashworthiness

Athough there are no federal standards regarding the spillage of electrolyte during a crash situation, proposed standards have been presented which would not permit any electrolyte spillage into the passenger compartment of the vehicle. This appears to be a reasonable goal and will be applied to the battery packaging in this program.

An additional goal in the packaging of the batteries is to position them, if possible, in a place where they will become involved early in the collision event in order to minimize the structural requirements of the supporting structure.

6.5.3.7 Handling Characteristics

The three most important packaging factors for consideration regarding the handling characteristics of the vehicle are the weight distribution, the polar moment of inertia, and the suspension type.

To assure that the final vehicle will have handling responses similar to other vehicles in its class, it is important that the car be front-weight biased. This will tend to result in a vehicle that is understeering in nature, which is the desired result.

The polar moment of inertia of the vehicle has a strong influence on its dynamic response character. High polar moments will result in a car that exhibits response times that are too long to produce a "stable feel." For this reason, every attempt will be made to minimize the polar moment of inertia.

The selection of suspension type also has a significant effect on the handling characteristics of the vehicle. Front suspensions should exhibit high camber coefficiets while rear-suspension systems should exhibit low camber coefficients. Both systems should be capable of providing roll and deflection understeer properties, if possible.

6.5.3.8 Serviceability

Access for service of the heat engine, control system, and battery must be maintained. One goal is to provide a means of servicing the battery without removing it from the vehicle. Battery replacement, when required, should not be a major maintenance project.

6.5.4 PRELIMINARY SCREENING

There are obviously a large number of vehicle packaging arrangements which are possible. To make the task of selection somewhat manageable, certain limitations were imposed on the preliminary packaging studies. Paramount among these limitations was acceptance of the Chevrolet Malibu seating arrangement. Secondly, only three basic drive arrangements were considered. These were front engine-motor/rear drive (F-R), front engine-motor/front drive (F-F), and rear engine-motor/rear drive (R-R), which represent all of the contemporary packages.

Starting from the above ground rules, it is clear that the design is really controlled by the location of the fuel tank and the position of the batteries.

6.5.4.1 Fuel System

There are four locations for fuel in modern passenger cars and although the shape of the fuel tank might vary, these may be classified with regard to their location.

Location No.	Location
F1	Front tunnel
F2	Below rear seat
F3	Over rear kick-up
F4	Between rear axle and rear bumper

Location F4, although widely used in currently produced American automobiles, was not considered a viable alternative due to the more stringent crash standards pending which would make it very difficult to satisfy the requirements of MVSS 301.

6.5.4.2 Battery Location

Of the many potential locations for the batteries, six were selected as the most likely locations for preliminary screening.

Location No.	Location
B1	Cowl
B2	Front backbone tunnel
B3	Rear tunnel
B4	Under rear seat
B5	Between rear wheels
B6	Behind rear wheels

6.5.5 PACKAGE SELECTION

In order to make a final selection from the twenty-four basic arrangements, a trade-off analysis was conducted with consideration of the following elements:

- Weight distribution
- Polar moment of inertia
- Suspension type
- Passenger compartment intrusion
- Crash structural requirements
- Battery spilling
- Utility
- Battery serviceability

- Heat engine serviceability
- Drive stability
- Proximity of controls to battery and motor
- Normal structure requirement

A trade-off chart was constructed utilizing the reference ICE vehicle as a base. Since many of the items listed above require more detailed design effort, or a subjective evaluation, judgments were made based on past experience. A grading system was established based on the merits of each arrangement when compared to the Reference Vehicle. If the item was rated the same, it was given a rating of '0;' if better, a rating of '1' was applied; if poorer, a rating of '-1' was given. The following sections discuss each item and the ratings applied.

6.5.5.1 Weight Distribution

The selection of the final package was based on a number of criteria. For the initial screening, the weight distribution was calculated for each of the possible candidate systems. Table 6-3 presents the results of this study. The numbers indicate the percentage of the vehicle curb weight carried by the front wheels. Arrangements marked n/a are impossible or nonfunctional placements and were not considered.

Based on the handling parameters discussed in Section 6.5.3.7, all arrangements which exhibit rear weight bias were eliminated from consideration. This includes all of the rear engine/motor-rear drive options and all but two of the front engine/rear drive packages.

6.5.5.2 Polar Moment of Inertia

A significant factor in the vehicle transient handling response characteristic is the polar moment of inertia. Should this factor grow too large, the response times will be excessively long regardless of the weight distribution or suspension characteristics. Polar moments of inertia were calculated for each of the arrangements not eliminated by the weight distribution criteria. The results are summarized in Table 6-4. Typical American passenger cars have polar moments of the order of 100,000 to 130,000 lb-ft^2. For the purposes of the trade-off analysis, values under 150,000 were rated 1, values under 190,000 were rated 0, while values over 190,000 were rated -1.

6.5.5.3 Suspension Type

The handling parameter considered was the suspension. If a solid axle could be utilized in the rear, the arrangement was scored as a 0 (same as the reference vehicle). If the arrangement demanded an independent rear suspension, it was scored a -1 due to the adverse roll camber and deflection steer coefficients typical of rear independent suspension.

Table 6-3

RESULTS OF WEIGHT DISTRIBUTION STUDY

F-F Arrangements

	F1	F2	F3
B1	0.660	0.650	0.648
B2	n/a	0.625	0.621
B3	n/a	n/a	0.524
B4	0.536	n/a	0.520
B5	0.503	0.491	0.488
B6	0.468	0.455	0.452

F-R Arrangements

	F1	F2	F3
B1	n/a	0.648	0.645
B2	n/a	n/a	n/a
B3	n/a	n/a	n/a
B4	n/a	n/a	n/a
B5	n/a	n/a	n/a
B6	n/a	0.452	0.449

R-R Arrangements

	F1	F2	F3
B1	0.447	0.435	n/a
B2	n/a	0.409	n/a
B3	n/a	n/a	n/a
B4	0.320	n/a	n/a
B5	n/a	n/a	n/a
B6	0.251	0.239	n/a

6.5.5.4 Passenger Compartment Intrusion

Passenger compartment intrusion is defined as a bump or lump in the passenger compartment which does not interfere with the seating package but imposes some comfort restriction, i.e., a transmission tunnel. Ratings were again assigned as relating to the Reference Vehicle. A '-1' rating was assigned to battery positions B2 and B3 while a '1' rating was assigned to all F-R configurations and to fuel position F1. A '+2' rating was given to all arrangements which resulted in less intrusion than the Reference Vehicle (with rear-wheel drive).

Table 6-4

RESULTS OF POLAR MOMENTS OF INERTIA STUDY

F-F Arrangements
Polar Moments (lb-ft^2)

	F1	F2	F3
B1	142700	159600	179200
B2	---	134800	143500
B3	---	---	205900
B4	196130	---	212400
B5	280400	---	---

F-R Arrangements
Polar Moments (lb-ft^2)

	F1	F2	F3
B1	---	165600	176700

6.5.5.5 Crash Structure Requirement

The requirement for crash structure is primarily dictated by the position of the batteries in the vehicle because they are the largest single, nonload-carrying mass. Highest ratings (minimum structural requirements) were given to battery position B2 since little or no additional structure would be required in the vehicle since the batteries are involved in the crash event early, and do not require structure to "ride down on." Location B1 was scored as '0' since it will require an improved cowl structure behind the batteries to prevent impingement upon the passenger compartment even though they do not require ride-down structure. Location B3 was also scored '0' since the structure is available in the mounting to provide ride-down. All other locations were scored -1 due to their severe ride-down structural requirements.

6.5.5.6 Battery Electrolyte Spillage

High marks were given to arrangements B3, B4, and B5 since the batteries would not be expected to be involved in the crash event and consequently would not rupture. A rating of '0' was assigned to B2 since, although the batteries will be involved in the crash event, they will be sheltered from the occupant compartment. A '-1' rating was given to arrangement B1 since the batteries will be involved in the crash event and must be protected or covered to prevent electrolyte spillage into the passenger compartment.

6.5.5.7 Utility

Utility ratings were based on the amount of free space available for future adaptation to station wagons, coupes, and other models. These same configurations will also permit maximum utilization of the baseline vehicle for luggage, etc.

6.5.5.8 Battery Serviceability

High ratings were given to arrangements which allowed battery service and inspection without removal from the vehicle. Lowest ratings were given in B2 and B4 which require battery removal from below for service or exchange.

6.5.5.9 Heat Engine Serviceability

Each arrangement, except the B1 series provides similar serviceability to the reference vehicle and were rated '0.' A '-1' rating was given to B1 since access to the heat engine is reduced.

6.5.5.10 Drive Stability

High ratings were given to front drive configurations because of their improved stability on low-coefficient surfaces and during regenerative braking.

6.5.5.11 Proximity of Controls

High ratings were given to arrangements in which the controls, battery, and drive motor were in close physical proximity. Arrangements which necessitated routing of cables twice along the length of the vehicle were given low ratings.

6.5.5.12 Normal Structural Requirements

High ratings were given to arrangements which lend themselves to simple structural paths. Arrangements which required battery removal from below were given '-1' ratings due to the difficulty in providing structural integrity around or under these large "holes" in the structure.

6.6 TRADE-OFF ANALYSIS

Table 6-5 summarizes the ratings given to the candidate layout configurations. Based on the chart, arrangements FFB1F2, FFB1F3, FFB2F2, and FRB1F2 were the most attractive having an accumulative rating of +2 or +3. Of the four prime contenders, arrangement FFB1F2 was selected as the preferred design and as the starting point for the preliminary design in Task 3. It is felt that the over-all advantage of FFB1F2 would have been even more dominant had the various design ranking factors been unequally weighted. For example, the total absence of any intrusion into the passenger compartment by the batteries (B1) and the ideal location of the fuel tank under the rear seat (F2) are very large

Table 6-5

CONFIGURATION RATING

Arrangement:	Polar Moment	Suspension	Passenger Compartment	Crash Structure	Battery Spill	Utility	Battery Service	Heat Engine Service	Drive Stability	Control Proximity	Normal Structure	Total Points
FFB1F1	1	0	0	0	-1	1	1	-1	1	-1	0	1
FFB1F2	0	0	1	0	-1	1	1	-1	1	1	0	3
FFB1F3	0	0	1	0	-1	0	1	-1	1	1	0	2
FFB2F2	1	0	-1	1	0	1	-1	0	1	1	-1	2
FFB2F3	1	0	-1	1	0	-1	-1	0	1	1	-1	0
FFB3F3	-1	-1	-1	0	1	-1	0	0	1	0	1	-1
FFB4F1	-1	0	0	-1	1	0	-1	0	1	0	-1	-2
FFB4F3	-1	0	1	-1	1	-1	-1	0	1	0	-1	-2
FFB5F1	-1	-1	0	-1	1	-1	1	0	1	-1	1	-1
FFB1F2	0	0	0	0	-1	1	1	0	0	1	0	2
FFB1F3	0	0	0	0	-1	-1	1	0	0	1	0	0

advantages for the FFB1F2 arrangement. The disadvantages (battery spill and heat engine service) of the selected arrangement are felt to be minor and can be avoided or overcome by good engineering and planning. The FRB1F2 arrangement was not selected because it is felt that front-wheel drive has inherent advantages over rear-wheel drive in the areas of overall packaging and vehicle handling. The United States suto industry is currently beginning to exploit these advantages.

Figure 6-12 illustrates the final package selected. The specifications are listed in Table 6-6.

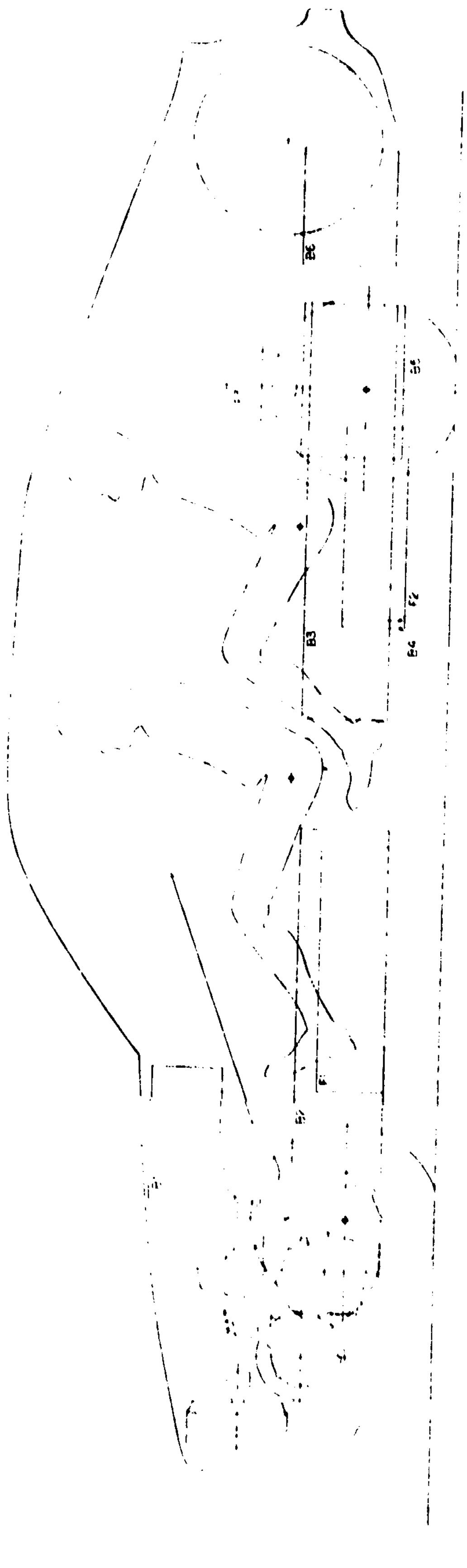


Figure 6-12. Packaging Composite

Table 6-6

SPECIFICATIONS FOR SELECTED PACKAGE

GENERAL		
Curb weight, kg (lb)	1754	(3860)
Weight distribution, F/R, %	62/38	
Gross vehicle weight, kg (lb)	2186	(4810)
GVW weight distribution F/R	53/47	
Wheelbase, cm (in.)	2743	(108)
Length, cm (in.)	4902	(193)
Width, cm (in.)	1816	(71.5)
Height, cm (in.)	1353	(53.3)
Ground clearance, cm (in.)	137	(5.4)
Overhang, F/R, cm (in.)	91.4/124.5	(36/49)
Trunk space, ℓ (ft^3)	481	(17)
Fuel capacity, ℓ (gal)	45	(12)
Seating capacity	5	
DRIVE		
Heat engine	gasoline or diesel	
Type	SOHC Inline 4	
Displacement, ℓ ($in.^3$)	1.6	(97)
BHP, hp	78	
Carburation (gasoline)	Fuel-injected	
Exhaust-emission control (gasoline)	Three-way catalyst	
ELECTRIC MOTOR		
Type	dc separately excited	
Frame	GE-2364	
Rated power	20 kW	
Transmission	Four-speed automatic shifted or steel-belt CVT	
CHASSIS AND BODY		
Layout	Front engine/front drive	
Body/frame	Unit steel	
Steering	Recirculating ball Hydraulic assist (close-centered)	
Front suspension	SLA independent	
Rear suspension	Solid axle	
Brakes	Hydraulic power assist	
ACCESSORIES		
Heater	Gasoline burner	
Air conditioner	Vapor compression Constant speed drive	

Section 7

CONTROL STRATEGY TRADE-OFF

Section 7

CONTROL STRATEGY TRADE-OFF

7.1 INTRODUCTION

The control strategy for operating the electric and heat engine drive systems separately and in combination is critical in meeting the design goals of the hybrid/electric vehicle. There are a multiplicity of possible control strategies. Even after a basic control approach has been selected, there are numerous alternative control options associated with the various operating modes of the vehicle. These options must be identified and evaluated. Development of the control strategy is a continuing effort and, at the present stage of the project, is being done using the HYVEC simulation program to test the various options. In this section of the report, the control strategies currently being used in HYVEC are discussed, and changes being considered for implementation during the Preliminary Design Task are identified. In those instances in which several control options were considered, the trade-offs involved in selecting the preferred option are discussed.

7.2 PRIMARY DRIVE SYSTEM SELECTION AND POWER SHARING

The key consideration in developing the control strategy for the hybrid/electric vehicle is the requirement that the vehicle use primarily electricity in urban driving when the state-of-charge of the battery permits. The electric drive system should be the primary drive system under those circumstances, and the heat engine would be used only to meet peak power demands that could not be met using the electric drive alone. In the present version of the HYVEC simulation program, the change from using the electric drive as primary to using the heat engine as primary is a function of vehicle speed, battery state-of-charge, and battery voltage. At low speeds the vehicle is operated using the electric drive alone, except when the battery state-of-charge is so low that it must be charged by the heat engine. Power demand permitting, the electric drive remains the primary drive system until the vehicle speed reaches VMODE, where

$$\text{VMODE} = \text{VMODEO}, \quad S \le 0.5$$

$$\text{VMODE} = \text{VMODEO}\ (1 - S^2), \quad S > .5$$

where

$$S = \frac{(\text{Ah})_{\text{used}}}{(\text{Ah})_{\text{cell capacity at avg current}}}$$

S is the indicator of battery state-of-charge and is simply the ratio of net ampere-hours used from the battery to the ampere-hour capacity (per cell) of the battery at the average discharge rate. VMODEO is selected such that the vehicle can be driven primarily as an electric vehicle below that speed and, in addition, such that for the EPA urban cycle most of the energy to drive the vehicle would be supplied by the battery. To date, most of the computer simulations of hybrid vehicles have been done using VMODEO equal to 50 km/hr (31 mph). When the battery is more than 50% discharged, VMODE is gradually decreased until at 80% depth of discharge, VMODE = 18 km/hr (11 mph). At vehicle speeds below VMODE, the engine is shut off, and it is not restarted until vehicle speed again exceeds VMODE or the power required exceeds that which can be supplied by the electric drive system. The maximum power that can be obtained from the electric drive is limited by the allowable armature current and the minimum battery voltage. If either limit is exceeded (armature current required is too high or battery voltage is too low), the heat engine is started and it supplies one-half of the power demanded to power the vehicle. The heat engine is shut off at speeds below VMODE when the power required drops below the rated power of the electric motor.

The heat engine is used to charge the battery only when the battery depth of discharge exceeds 80%, and when this occurs the battery is only recharged until the depth-of-discharge is reduced to 70%. Charging of the battery by the heat engine is restricted to this narrow range to avoid returning to the garage with significant battery charge after driving many miles (>75 mi) in the urban area.

The control strategy discussed thus far is that for urban driving, the intent is to use primarily electricity and as little gasoline as possible. For long, intercity trips, it is recognized at the outset that the vehicle will be operated primarily on gasoline, and that for highway travel, the primary function of the electric drive is to load-level the heat engine. This is done by using the electric drive only for low-speed accelerations and for passing at high speeds when the power required is greater than that which can be supplied by the heat engine. In the HYVEC simulations on the Environmental Protection Agency highway cycle, a value of VMODE equal to 60 kW/hr (37 mph) has been used. This results in a range of about 400 mi for the batteries on the Environmental Protection Agency highway cycle.

Considerable work remains to be done in developing the control strategy for the hybrid/electric vehicle and the implementation of the strategy using a microprocessor. Much of this work will be done during the Preliminary Design Task. Particular attention will be given to blending the power inputs from the electric and heat engine drives when both systems are required. In addition, work is needed to determine the anticipation time required by the heat engine or electric drive systems when they are the secondary drive systems (in a stand-by condition). As indicated in Figure 7-1, there are losses associated with maintaining the secondary

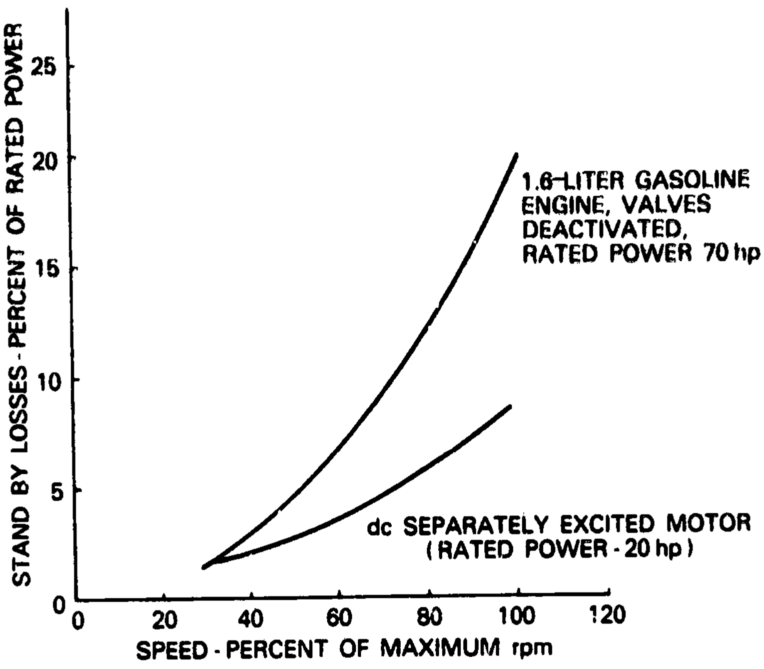


Figure 7-1. Standby Losses of the Heat Engine and dc Electric Motor as a Function of Speed

drive system in an advanced state-of readiness (i.e., up-to-speed and turning, but not energized). These losses are particularly large in the case of the heat engine. The rotating losses are not presently included in HYVEC, but they have been identified and will by included in the near future. An important control strategy trade-off is the extent to which the secondary drive system is maintained in a condition for near-instantaneous use.

Another area in which further development of the control strategy is required is the relationship between VMODE and power demand. A greater fraction of gasoline can be saved if the electric drive system is maintained as the primary drive in urban use for speeds above VMODE if the power required to power the vehicle is less than the continuous rated power of the electric motor. This will reduce the effective electric (battery) range of the hybrid vehicle and necessitate charging the batteries by the heat engine sooner than otherwise would be the case. The advantages/disadvantages of using the electric drive more of the time for vehicle speeds greater than 30 mph in urban driving requires additional analysis using HYVEC. These trade-offs will be made during the continuing control strategy development.

7.3 SHIFT LOGIC

The use of an automatically shifted gearbox or CVT in the drive-line is particularly needed when the heat engine is the primary power source. The transmission is also needed to provide adequate gradability using the electric drive and to optimize its operation. The rotational speed requirements of the electric and heat engine drive systems are quite different. In general, it is desirable to operate the electric motor at relatively high rpm (above base speed) and to operate the heat engine at as low an rpm as possible consistent with meeting the power requirements. Hence, the shifting logic or speed ratio control of the CVT must be different depending on whether the electric drive or heat engine is the primary drive system. At the present time, the shifting logic favors whichever power source is operating and favors the electric motor if both the motor and engine are operating. For the electric motor, shifting to the next higher gear occurs when the motor speed reaches 85% of the maximum rpm. When the heat engine is the primary drive, shifting occurs when the engine rpm reaches 35% of the maximum rpm unless that is precluded by the power demanded. This engine shifting logic is intended to maximize engine efficiency. At the present time, the shifting logic depends primarily on vehicle speed but is subject to power and electrical system constraints. The latter are particularly important during regenerative braking and acceleration using battery switching as discussed in later sections.

Further optimization of the shifting logic (or speed ratio control in the case of a CVT) will be undertaken in the preliminary design phase. The major trade-offs are between system efficiency, decision-making complexity, and the need for repeated transmission shifts. Evaluation of these trade-offs will require detailed design and simulation and possibly even testing of the power train in a vehicle during Phase II of the program.

7.4 REGENERATIVE BRAKING

The control strategy required to implement regenerative braking is rather simple. First, the motor/generator must be at or brought to a sufficiently high rpm so that it can generate the voltage required to charge the battery at the current corresponding to the needed braking torque. If the electric drive is the primary drive system, the rpm requirement is likely to be satisfied. When the heat engine is the primary drive system, it is necessary to maintain the motor turning at a relatively high rpm or the opportunity for regenerative braking probably cannot be utilized. The losses associated with the motor turning in an unenergized state are tolerable (see Figure 7-1) in urban driving and such is assumed in the present HYVEC program. The battery charging current associated with regenerative braking is relatively high (> 150 A), but the battery can accept the current at a voltage consistent with the motor/generator output except when the battery is near full charge (e.g., $S < 0.05$). It might be desirable to provide dynamic braking (dissipation of current in a resistor) when the charging current is higher than the battery can accept. Downshifting of the transmission is required during braking to maintain a motor/generator output voltage consistent with the battery charging characteristics which, as just noted, are dependent on the state-of-charge. The use of dynamic braking may also prove to be advantageous in simplifying the shifting logic during braking.

In highway driving, the potential energy recovery by regenerative braking is small. Hence, the losses associated with maintaining a high motor rpm may not be tolerable when the heat engine is the primary drive system unless such is required to achieve adequate passing capability. Since longer hesitation is tolerable for initiating a passing than is tolerable for a stopping maneuver, it is likely that further analysis will indicate regenerative braking at highway speeds is not worth the associated losses.

7.5 BATTERY SWITCHING

Without armature chopper control, the electric motor cannot operate below its base speed, which depends both on motor design and battery voltage. Base speed increases proportionally to battery voltage. Since relatively high voltage is desirable at high motor loads and low voltage is desirable at low loads to reduce motor idle speed, battery switching was studied as a means of meeting these conflicting requirements. Hence, a strategy is required to control under what conditions the batteries are switched from a parallel to a series connection, which, in effect, doubles the voltage applied to the motor. This also doubles the base speed of the motor. In the present version of the HYVEC program, the battery switching takes place when the motor rpm at the lower battery voltage (parallel connection) exceeds the base speed corresponding to the series connection of the batteries. It is more efficient to perform the battery switching as soon as the motor rpm is high enough so that the power required can be supplied by the motor connected to the battery in the series configuration. In this way, the motor flux is increased and the motor current is lowered for the same power output. As a result, I^2R losses are at a minimum during the vehicle acceleration. This modification of the battery switching control strategy will be studied during the Preliminary Design task. Since it is desirable during battery switching to have the motor at the highest rpm possible, the transmission will be in the lowest gear during and after the battery switching and until the motor is well past base speed rpm.

7.6 ACCESSORIES

One of the most complex aspects of developing a control strategy for the hybrid vehicle is that of providing for the required vehicle accessories, including heater/defroster, air-conditioning, power steering, lights, etc. Some of these accessories, such as the heating and air-conditioning, require such large amounts of energy (see Section 6.4) that using electricity to provide them seems to be precluded. Hence, it appears that it will be necessary to use the engine fuel to meet unusually high accessory loads either in an auxiliary gasoline burner or by idling the engine. Normal accessory loads, such as lights, radio, hydraulics, etc., can be provided even when the vehicle is at rest by idling the electric motor. It seems necessary that the operating strategy of the power train be dependent on whether the air-conditioner is operating or is off. This is especially true during the initial cool-down period after the vehicle has been soaking for a long period in the heat. It may be possible to run the air-conditioner from the electric motor after the initial cool-down period without drastically reducing the electric range of the hybrid vehicle. These possible means of providing air-conditioning have not as yet been compared from an over-all gasoline/total energy point of view, but this will be done in the next task. All of these control options will require increased capacity in the system microprocessor.

Heating and defrosting will be provided by a combination of an auxiliary gasoline burner and waste heat from the engine. No special control options are contemplated in this regard because it is most efficient to provide heat directly using the burner unless waste heat is naturally available from the engine.

In general, every attempt is being made to minimize the impact on the control strategy and energy use (both fuel and electricity) due to the required accessories. Close-centered hydraulic systems using a hydraulic accumulator to store energy are being considered as one means of uncoupling the accessory requirements from the control strategy and to minimize electrical energy use at rest.

Section 8

COMPONENT SELECTION AND SIZING TRADE-OFF FOR VARIOUS DRIVING CYCLES

Section 8

COMPONENT SELECTION AND SIZING TRADE-OFF FOR VARIOUS DRIVING CYCLES

8.1 INTRODUCTION

Candidate components and power train configurations for hybrid/electric vehicles were evaluated and compared in Section 5 based on vehicle synthesis calculations. The most attractive of those components/configurations have been analyzed further in considerable detail using the HYVEC second-by-second vehicle simulation program. This has permitted a detailed comparison of the candidate hybrid power trains in terms of characteristics, such as fuel economy, total energy usage, emissions, acceleration, performance, etc., which can only be determined from second-by-second simulations over specific driving cycles.

The hybrid power train components/configurations studied using HYVEC are summarized in Tables 8-1, 8-2, and 8-2a. It is clear from the tables that an extensive series of runs was made permitting the comparison of the most attractive candidate hybrid power train approaches. In some cases it was possible to make clear-cut decisions/conclusions from the results, while in others the calculations indicated the trade-offs which must be faced in selecting one component rather than another. These decisions/trade-offs will be highlighted during the course of the discussions of the HYVEC results. All the hybrid vehicle power train evaluation calculations were made using the same vehicle weight and drag values (see Table 8-1).

Table 8-1

SUMMARY OF VEHICLE CHARACTERISTICS USED IN THE HYBRID POWER TRAIN EVALUATION CALCULATIONS

Inertia Weight	- 1818 kg (4000 lb)
Drag Coefficient	- 0.40
Frontal Area	- 2 m^2 (21.5 ft^2)
Rolling Resistance	- 0.011 lb/lb
Wheel Diameter	- 0.58 m (22.8 in.)
Wheel Inertia	- 1.1 $kg\text{-}m^2$
Axle Ratio	- 3.3
Differential Efficiency	- 96%

Table 8-2

SUMMARY OF HYBRID POWER TRAIN CONFIGURATIONS AND OPERATING MODES CONSIDERED

Power Train Configuration

- Parallel

Engine Types

- Gasoline
- Diesel

Transmission

- Automatically Shifted; Four-speed Synchromesh.

Electric Drive System

- Direct Current Separately Excited
  - Armature and Field Control
  - Field Control and Battery Switching
  - With/Without Regeneration

Batteries

- Lead-Acid (700 lb)
- Ni - Zn (500 lbs and 700 lb)
- Ni - Fe (700 lb)

Control Strategy

- Varied VMODE
- Engine On/Off versus Engine Idling

Driving Cycles

- EPA Urban (Transient and Stabilized Parts)
- EPA Urban (Stabilized Part)
- EPA Highway
- SAE J227a, Schedule B

8.2 HYVEC PROGRAM VALIDATION

The HYVEC program was developed to simulate the hybrid (heat engine and electric) vehicle, but it can treat the conventional ICE vehicle and the all-electric vehicle as special cases. It was, of course, of interest to utilize the program for these special cases for which other simulation programs and test data are available. Calculations were made using HYVEC for the Volkswagen Rabbit (gasoline and diesel), Chevrolet Malibu, Audi 5000 (gasoline and diesel), and the DOE-GE Near-Term Electric Vehicle. The results of the calculations are given in Tables 8-3 and 8-4.

Table 8-2a

HYBRID VEHICLE CHARACTERISTICS
FIXED FOR HYVEC CALCULATIONS

Inertia Weight	-	1818 kg (4000 lb)
Frontal Area	-	2 m^2
Drag Coefficient	-	0.40
Tire Rolling Resistance	-	0.011 kg/kg
Wheel Diameter	-	0.58 m
Wheel Inertia	-	1.10 kg-m^2
Axle Ratio	-	3.30

Table 8-3

FUEL ECONOMY AND ACCELERATION RESULTS
FOR SELECTED ICE VEHICLES*

			Fuel Economy (mpg)		Acceleration (S)	
Vehicle	Engine Type	Horsepower	City A (B)†	Highway A (B)†	0-50 Km/h	0-100 Km/h
VW Rabbit	Gasoline	72	24.3 (25)	33.1 (38)	4.5	14.6
VW Rabit	Diesel	70	35.8	42.8	3.9	13.2
Chev. Malibu	Gasoline	95	22.2 (21)	28.9 (29)	5.4	18.5
Audi 5000	Gasoline	103	18.2 (16)	27.1 (22)	4.3	14.3
Audi 5000	Diesel	70	33.0	43.3	6.1	19.5

*All vehicles utilized four-speed manual transmissions.
†A-calculated using HYVEC; B - EPA published fuel economy for 1978.

For the conventional ICE vehicles, if fuel economy results were available from EPA, they are noted in Table 8-3. In general, the HYVEC results for the conventional ICE vehicles are in good agreement with EPA values. In some instances, the HYVEC predictions are very close to those obtained by EPA, while in others the differences are significant. No effort was made to determine the reasons for the differences in particular cases. It was felt, however, that the results obtained did validate the HYVEC program for heat-engine-powered vehicles.

The results obtained using HYVEC for the DOE/GE Near-Term Electric Vehicle are given in Table 8-4. Results are shown for the SAE J227a B and D cycles and the stabilized portion of the EPA urban cycle. The same velocity versus time schedule was used in the HYVEC calculations for the SAE J227a D cycle as has been used by General Electric for the Near-Term Electric Vehicle Program range projections. As indicated in Table 8-4, the range predictions from the two General Electric programs are almost identical. Both

Table 8-4

SIMULATION RESULTS FOR THE DOE/GE NEAR-TERM ELECTRIC VEHICLE (W_V = 3750 lbs)

Cycle	kWh/mi	Range (mi) A	Range (mi) (B)*	Acceleration Time(s) 0-30 mph
SAE J227 D (Constant Battery Current Acceleration)	0.216	71	(71)	10 (9)
SAE J227 B	-	98	-	- -
EPA Urban (Stabilized Segment)	0.278	61	-	- -

*A-HYVEC; B-GE program developed for Near-Term Electric vehicle

programs utilize essentially the same motor, controller, and battery models, but the two programs are completely separate and share no subroutines.

8.3 ELECTRIC DRIVE SYSTEM OPTIONS

Calculations were made for several electric drive system options. All systems considered utilized a dc separately excited motor/generator. The options considered were:

- Armature and field control
- Field control with battery switching
- With and without regenerative braking

Results are shown in Figures 8-1 through 8-4 for hybrid power trains using a gasoline engine and lead-acid batteries (700 lb). Figures 8-1 through 8-3 indicate that the effect of electric drive-system control (armature and field control or field control with battery switching) on energy usage by the hybrid vehicle is quite small. As might be expected, the utilization of armature and field control permits a greater fraction of the power requirements to be met by using electricity and, thus, a slightly higher fuel economy and fraction of gasoline is saved, but there is also a slightly faster depletion of the battery. It has been concluded, however, that from an energy usage standpoint (fuel and electricity) field control with battery switching is a satisfactory means of controlling the dc electric drive system.

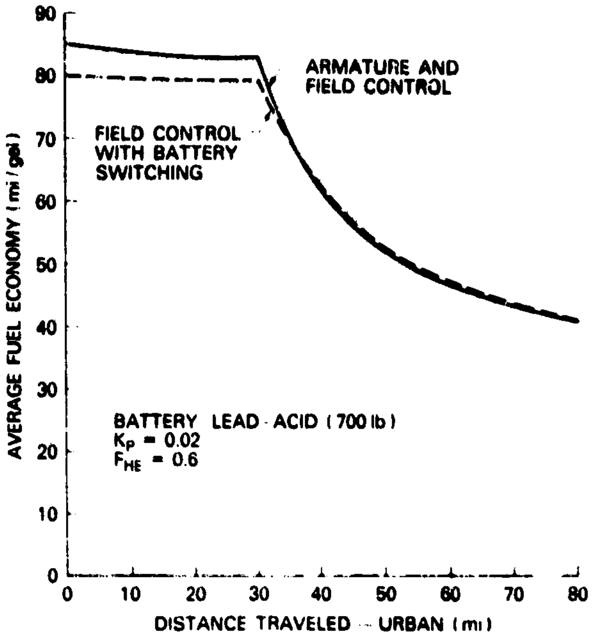


Figure 8-1. Effect of Electric Drive Control on Average Fuel Economy

The effect of regenerative braking on battery state of discharge and fuel economy is shown in Figure 8-4. It is seen from the figure that regenerative braking has a significant effect on the effective electric range of the hybrid vehicle for the EPA urban cycle. In the present regenerative braking calculations, it is assumed that the battery cannot accept regeneration currents until it is discharged 10% and thereafter it can accept all the regeneration currents as long as the motor/generator can provide the required charging voltage. As indicated in Figure 8-4, the need to recharge the battery from the heat engine is delayed beyond 75 mi of urban travel by using regenerative braking. In addition, the average fuel economy of the hybrid vehicle is significantly increased at ranges beyond 30 mi by regeneration. It seems clear from Figure 8-4 that the use of regenerative braking in the hybrid vehicle is justified.

8.4 HEAT ENGINE SELECTION AND SIZING

Hybrid vehicle calculations have been made for both gasoline and diesel engines. Most of the calculations were made for a 73-hp heat engine (K_p = 0.022) with the heat engine providing 60% of the peak power for the vehicle (F_{HE} = 0.6). Some calculations were made utilizing a smaller heat engine (45 to 50 hp) in combination with the same size electric motor (K_p = 0.0185) as used with the larger heat engine. This latter combination of electric motor and heat engine yielded the same all-electric performance but a degradation in performance of the hybrid vehicle at high speeds.

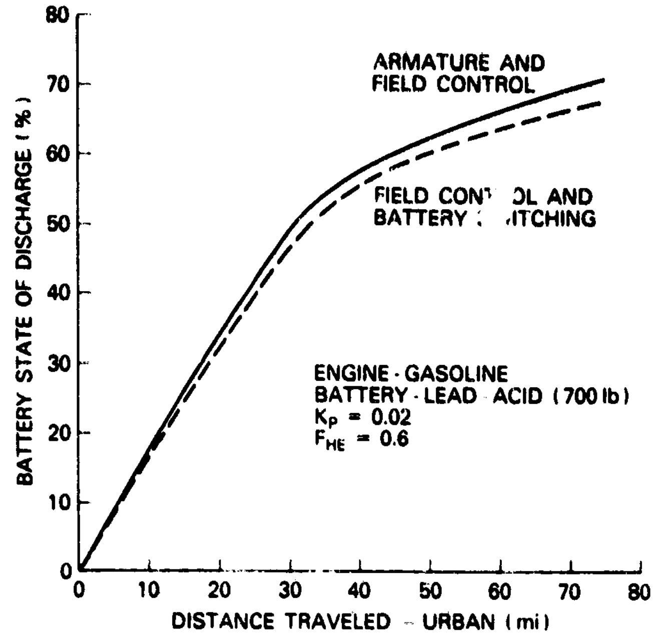


Figure 8-2. Effect of Electric Drive Control on Battery State of Charge

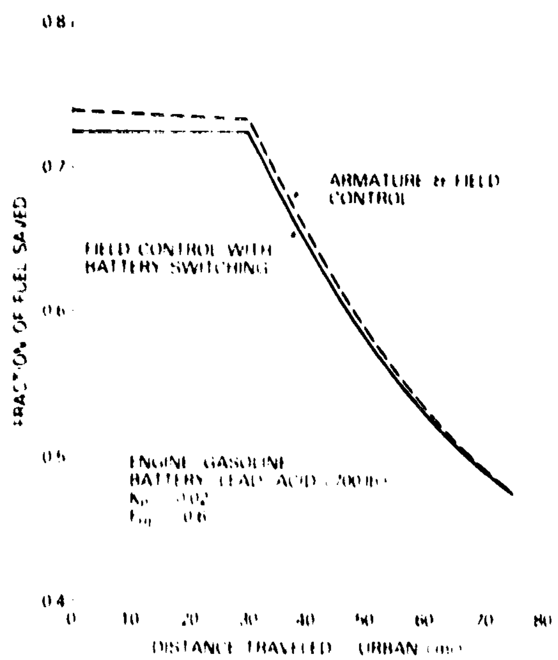


Figure 8-3. Effect of Electric Drive Control on Fraction of Fuel Saved

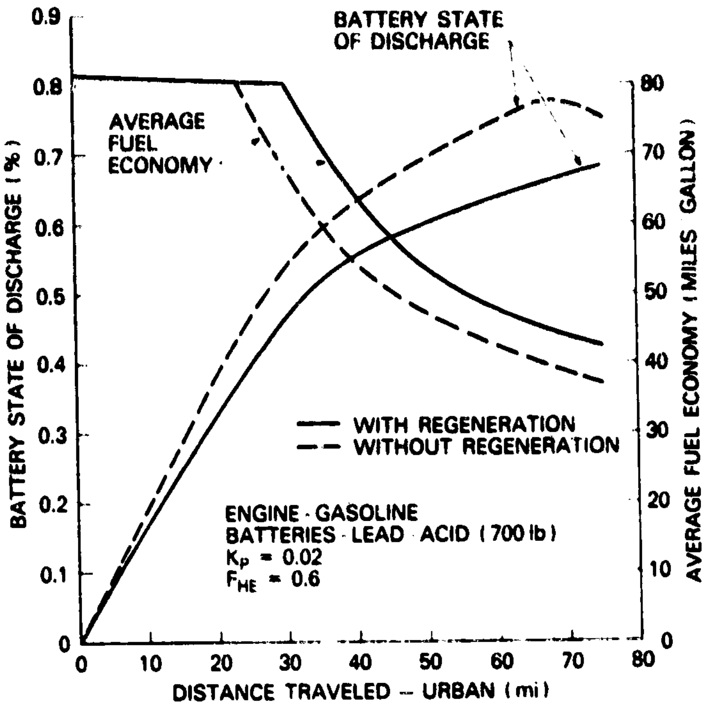


Figure 8-4. Effect of Regenerative on the Battery State of Discharge and Average Fuel Economy

Characteristics of hybrid vehicles using gasoline and diesel engines of the same peak power (73 hp) are shown in Figures 8-5 through 8-9. The comparison of the average fuel economy of the gasoline- and diesel-engine-powered hybrid vehicles is given in Figure 8-5. The diesel engine yields a higher fuel economy for all ranges with the advantage of the diesel being 25% for ranges less than 30 mi increasing to about 35% at 75 mi. The fraction of fuel saved and the battery state of discharge as a function of urban distance traveled are shown in Figures 8-6 and 8-7. For both parameters, the differences between the gasoline and diesel engine cases are quite small indicating that the dominant factor is the basically superior brake specific fuel consumption (bsfc) of the diesel engine. The total energy usage (fuel used by the engine plus that required to generate the electricity at the power plant) per mile is shown in Figure 8-8 as a function of urban travel. The advantage of the diesel-engine-powered hybrid is significantly reduced because the higher energy content (per gallon) of the diesel fuel is included in this calculation. For ranges less than 30 mi, the total energy advantage of the diesel-powered hybrid vehicle is about 6%, and at 75 mi, the advantage of the diesel is about 10%. Also it is of interest to note that hybrid operation benefits the gasoline-powered vehicle to a greater extent relatively than the diesel-powered vehicle.

The emissions (hydrocarbon, carbon monoxide, NO_x particulates) of the gasoline- and diesel-powered hybrid vehicles are compared in

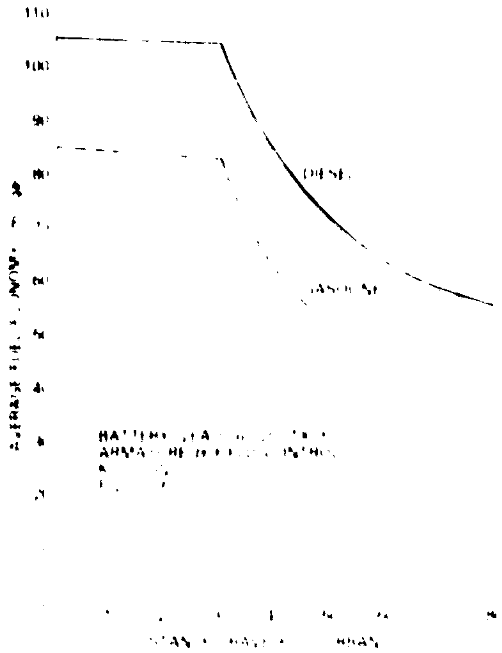


Figure 8-5. Effect of Heat Engine Type on Average Fuel Economy

Figure 8-9. In all cases the emissions (gm/mi) are unchanged during the electric range of the hybrid vehicle and increase at greater distances as the engine takes on a greater share of powering the vehicle. The gasoline-powered hybrid vehicle utilized a three-way catalyst; the diesel-powered hybrid had no exhaust gas treatment. Both the gasoline/hybrid and the diesel/powered hybrids met the most stringent carbon monoxide and hydrocarbon standards proposed to date (i.e. 0.4 gm/mi hydrocarbon and 3.4 gm/mi CO) out to ranges of 75 mi. The situation relative to NO_x emissions is more complicated. Both the gasoline/hybrid and the diesel/hybrid vehicles met the 1985 Federal standard of 1.0 gm/mi for ranges up to 75 mi, but neither met the proposed California or Federal Research Standard of 0.4 gm/mi even during the electric primary operating mode of the hybrid vehicle. It seems likely, however, that hybrid vehicles using both types of heat engines could meet the 0.4 gm/mi NO_x standard in the first 30 mi of operation with additional tuning and development. At greater ranges, meeting the 0.4 gm/mi NO_x standard becomes more difficult. The gasoline/hybrid vehicle using EGR and a properly sized three-way catalyst could probably meet the 0.4 gm/mi NO_x standard for all ranges, but the likelihood of the diesel/hybrid meeting the same NO_x standard is much less because that would require significant reductions in the untreated NO_x emissions from the diesel engine. Such reductions would require basic changes in the engine design and combustion process itself.

The diesel/hybrid vehicle has the special problem of particulate emissions. The vehicle simulation results shown in Figure 8-9 indicate a projected particulate emission of about 0.16 gm/mi for the first 30 mi and an average emission of about 0.33 gm/mi for the first 75 mi. The Environmental Protection Agency (EPA)

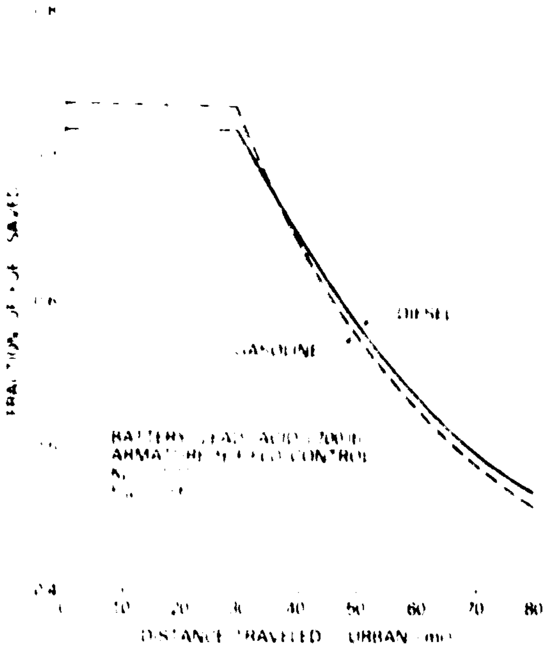


Figure 8-6. Effect of Heat Engine Type on Fuel Saved

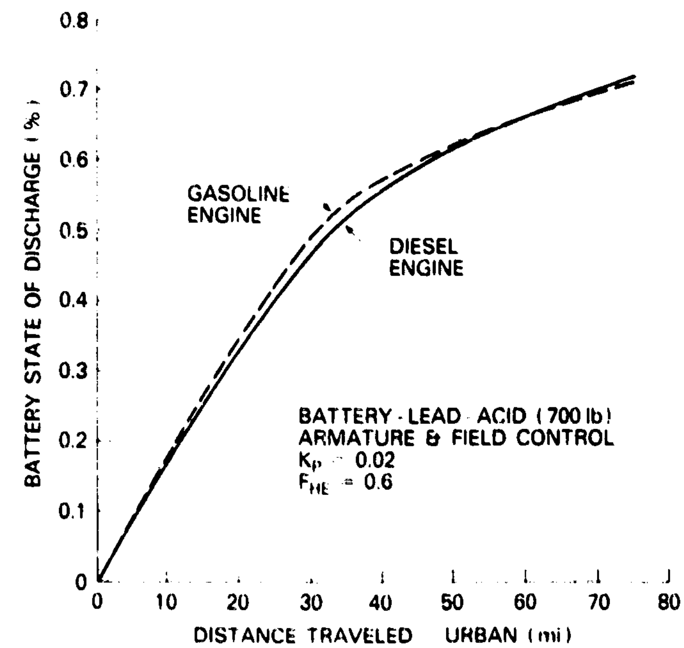


Figure 8-7. Effect of Heat Engine Type on Battery Discharge

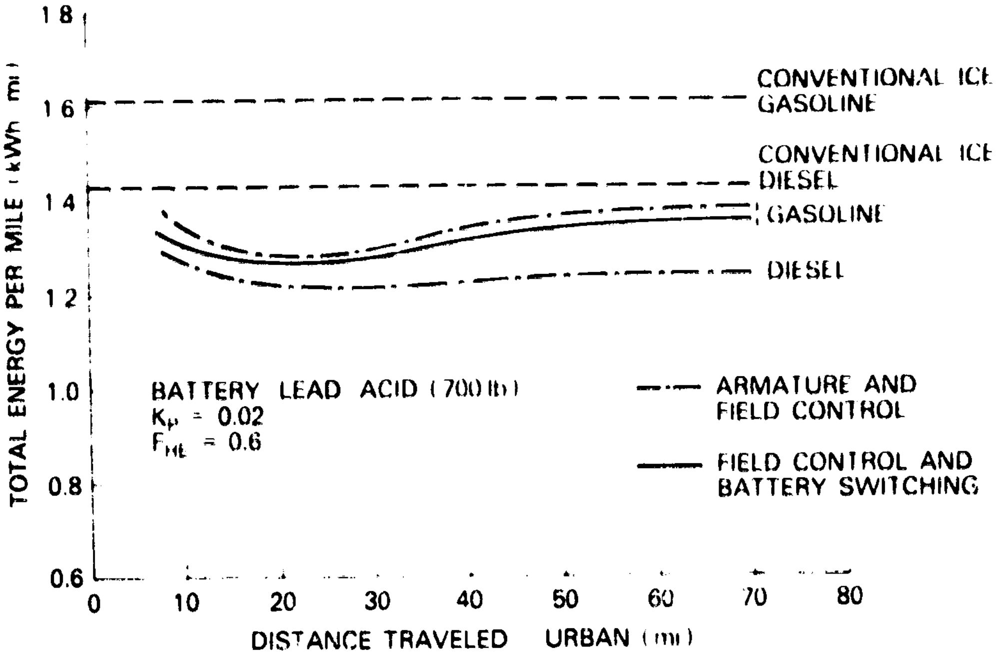


Figure 8-8. Effect of Heat Engine Type on Total Energy Expended

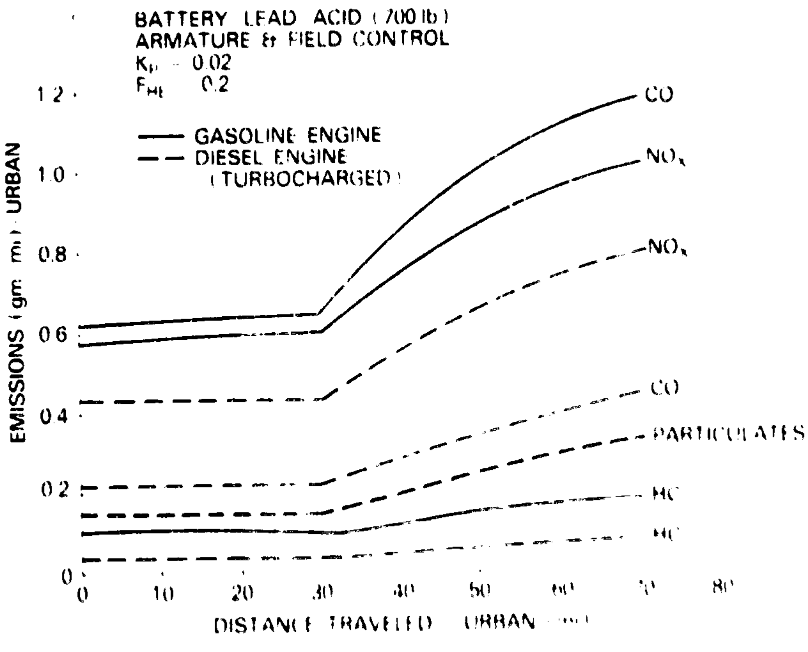


Figure 8-9. Emissions of the Gasoline and Diesel Engine-Powered Hybrid Vehicle in Urban Driving

has proposed the following particulate emissions standards: 0.6 gm/mi in 1981, 0.2 gm/mi in 1983. These proposed particulate standards are currently the subject of EPA hearings and are being vigorously contested by the U.S. and foreign auto industries who are claiming that they connot be met using existing technology. The results shown in Figure 8-9 indicate that this is also true for the diesel/hybrid vehicle with respect to the proposed 1983 standard of 0.2 gm/mile.

The situation regarding the use of the gasoline and diesel engine in a hybrid vehicle is very similar to that regarding their relative attractiveness in conventional ICE vehicles. As in the conventional vehicle, the diesel engine in the hybrid vehicle yields an improvement in fuel economy of 25 to 35% compared with the gasoline engine. The problems with the diesel engine are the difficulties in reducing NO_x emissions below 1.0 gm/mi and meeting the proposed EPA particulate standards of 0.2 gm/mi in 1983. The diesel engine also suffers from hard starting when cold. This could require idling the diesel engine until it is warmed-up even during the electric primary operating mode. It seems clear that from an emissions point of view and for operating in the on/off mode, the gasoline engine has clear advantages over the diesel engine. Whether these advantages more than compensate for its lower fuel economy is difficult to assess in light of the uncertainty concerning NO_x and particulate emission standards which will be in effect in 1985. The gasoline engine has been used as the prime heat engine candidate thus far in the present study because of the cited disadvantages of the diesel engine, but the option of using the diesel engine has been left open, and space is available for its inclusion in the power train of the vehicle during the Preliminary Design Task.

Vehicle simulation calculations were also made using smaller heat engines in the hybrid vehicle. The engines included, all available from Volkswagen, are listed below:

- Gasoline
  - 1.6 ℓ, 73 hp
  - 1.3 ℓ, 60 hp
- Diesel
  - 1.6 ℓ, turbocharged, 73 hp
  - 1.6 ℓ, naturally aspirated, 48 hp

The same 36 kW (peak rating) dc electric motor was used in all the power train combinations simulated. The results of the calculations are given in Figures 8-10 through 8-13. The fuel economy in urban driving using the various engines is given in Figure 8-10 and for highway driving in Table 8-5. In all cases the use of the smaller engine yields an improvement in fuel economy. Also, as expected (Figures 8-11 and 8-12), the battery is depleted slightly more rapidly, and the heat engine contributes a smaller fraction of the total energy to power the vehicle when smaller engines are used. The emissions for the turbocharged (73 hp) and

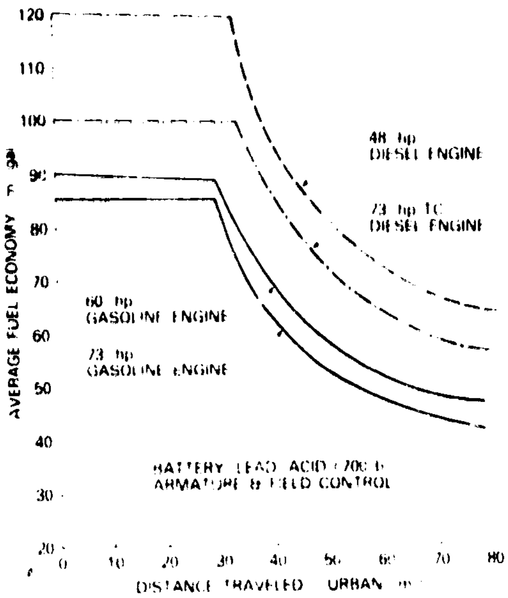


Figure 8-10. Effect of Heat Engines Type and Rating on Average Fuel Economy

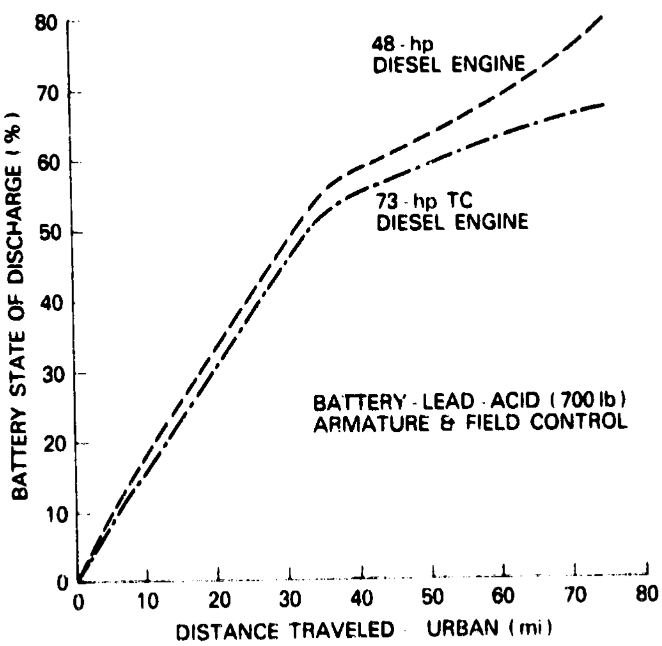


Figure 8-11. Effect of Heat Engine Rating on Battery State of Discharge

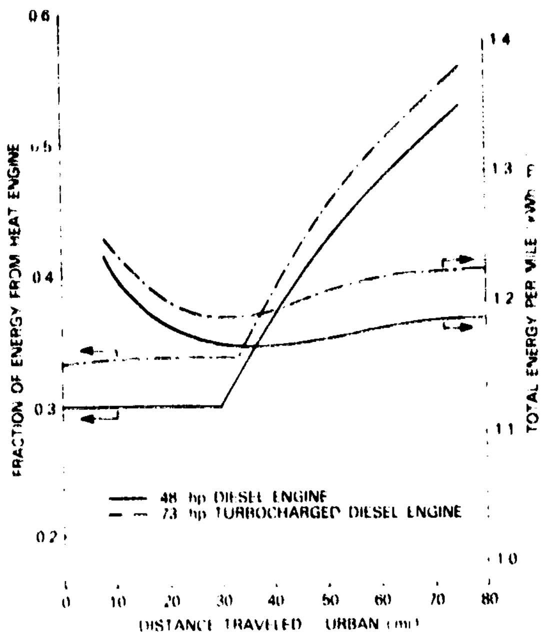


Figure 8-12. Effect of Heat Engine Rating on Total Energy and Heat Engine Fraction

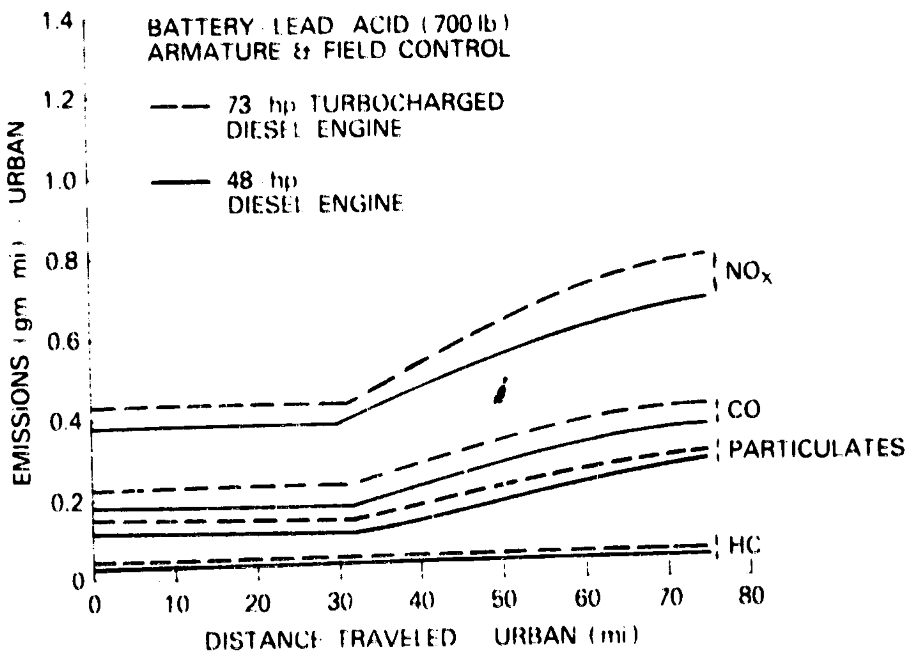


Figure 8-13. Effect of Heat Engine Rating on Emissions

naturally aspirated (48 hp) diesel engines are compared in Figure 8-13. The smaller engine yielded slightly lower emissions for all pollutants.

The disadvantage of using a smaller engine is that the acceleration performance of the hybrid vehicle is reduced. The minimum design goal set by the Jet Propulsion Laboratory in this regard is 0 to 60 mph in 15.5 s. The calculated acceleration time using the various engines in the hybrid power train are given in Table 8-6. The design goal is marginally met using the smaller engine when the batteries are fully charged. The design goal would not be met when the batteries are partially discharged. For this reason and because of the desire to have available additional power for recharging the batteries on the road when that becomes necessary, the Preliminary Design Task uses the larger 73-hp engines. The exterior profiles of all the engines are essentially the same (all use the same basic engine block), and those space requirements are not a key consideration.

Table 8-5

HIGHWAY DRIVING FUEL ECONOMY
USING VARIOUS HEAT ENGINES

Engine	Horse-Power	mpg	Fraction of Energy from Engine (%)	Change of State-of-Discharge per 100 mi (%)
Gasoline	73	32	96	10
Gasoline	60	36.5	94	15
Diesel	73	39	96	10
Diesel	48	45.9	94	15

Table 8-6

ACCELERATION TIMES USING VARIOUS
SIZE GASOLINE AND DIESEL ENGINES

Engine	Horse-Power	Acceleration Time (s)*	
		0-30 mph	0-60 mph
Gasoline	73	3.5	12.1
Gasoline	60	4.6	15.1
Diesel	73	3.9	12.5
Diesel	48	4.8	15.8

*Batteries fully charged

8.5 BATTERY SIZING AND SELECTION

Hybrid vehicle simulation calculations have been made using lead-acid, Ni-Zn, and Ni-Fe batteries. The battery cell capacity (AH) was sized so that the resultant battery pack weighed 700 lb or 500 lb for a nominal 108 V system. The battery cell/module characteristics used in the simulation calculations are given in Table 8-7. The results of the calculations are presented in Figures 8-14 through 8-19. The results using lead-acid and Ni-Zn batteries will be discussed together as both of those battery types have sufficiently high power density (i.e., low internal resistance) so that the battery is not the limiting factor in providing power from the electric drive system. It was found that even using 700 lb of Ni-Fe batteries, the battery pack was the power-limiting factor, and the behavior of the hybrid power train using Ni-Fe batteries was significantly different from that using the higher power density batteries.

Table 8-7

BATTERY CHARACTERISTICS USED IN HYBRID VEHICLE SIMULATION CALCULATIONS

Battery Type	Battery Weight (lbs)	Open-Circuit Cell Voltage	Cell Ah Capacity	Number of Cells in Module	Number of Cells
Lead-Acid	700	2.10	110	3	54
Ni-Zn	700	1.65	180	5	70
Ni-Zn	500	1.65	129	5	70
Ni-Fe	700	1.15	210	10	100
Ni-Fe	500	1.15	150	10	100

The battery state of discharge using lead-acid and Ni-Zn batteries in the hybrid vehicle is shown in Figure 8-14 as a function of miles traveled in urban driving. Results are given for both 500 lb and 700 lb of Ni-Zn batteries. The rate of depletion of the smaller Ni-Zn battery is slightly slower than that for the 700-lb lead-acid battery. This is to be expected as the 500-lb Ni-Zn battery stores 13.75 kWh of energy compared with 12.25 kWh for the lead-acid battery. The fuel economy (mpg) of the hybrid vehicle in urban driving is given in Figure 8-15.

The major effect of the different battery packs is to extend the electric range of the hybrid vehicle and, thus, its high electric primary fuel economy to longer distances. For example, the electric range using 700 lb of Ni-Zn batteries is 52 mi compared with 30 mi for 700 lb of lead-acid batteries. The fraction of gasoline saved using the various battery packs is given in Figure 8-16. As expected, the Ni-Zn batteries permitted the saving of a greater fraction of gasoline at ranges in excess of 30 mi.

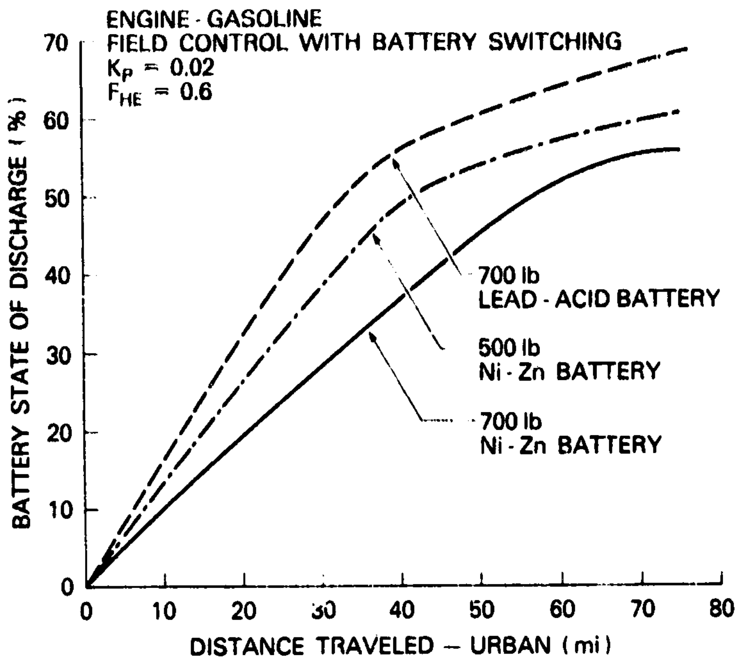


Figure 8-14. Effect of Battery Type on Battery State of Discharge

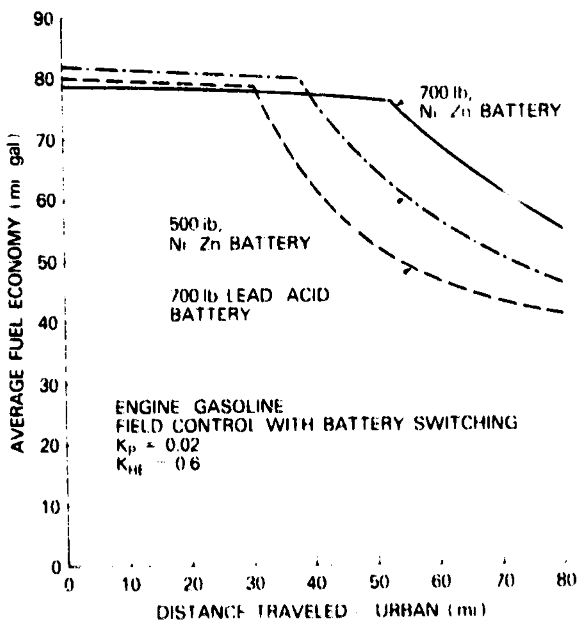


Figure 8-15. Effect of Battery Type on Average Fuel Economy

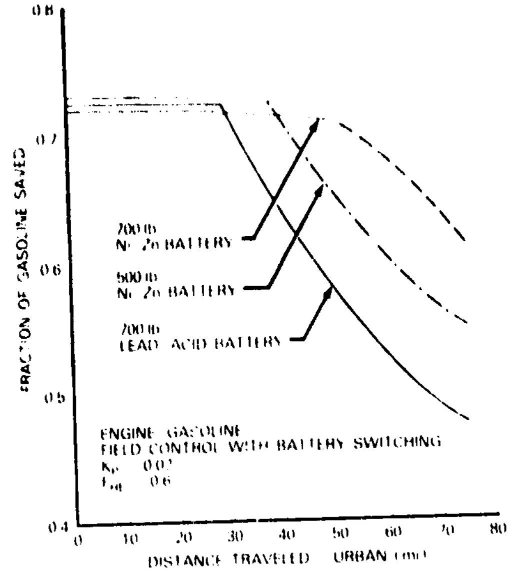


Figure 8-16. Effect of Battery Type on Fuel Saved

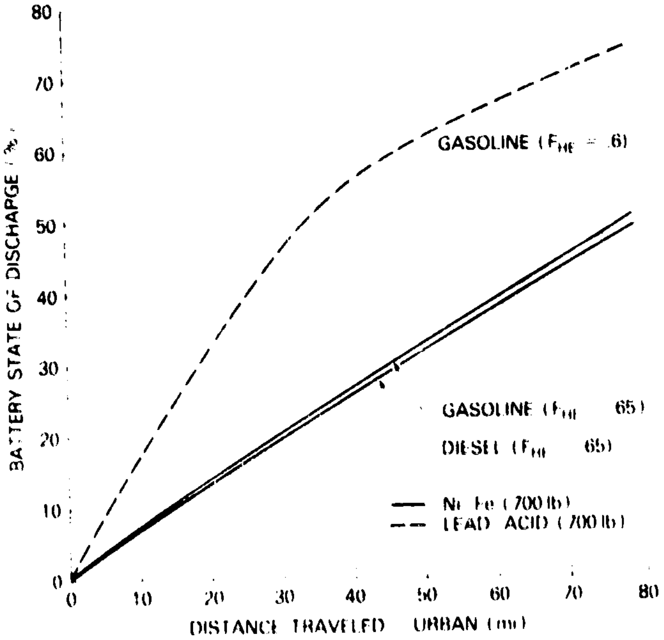


Figure 8-17. Comparison of the Change of Battery State-of-Charge with Distance for Hybrid Vehicles Using Ni-Fe and Lead-Acid Batteries

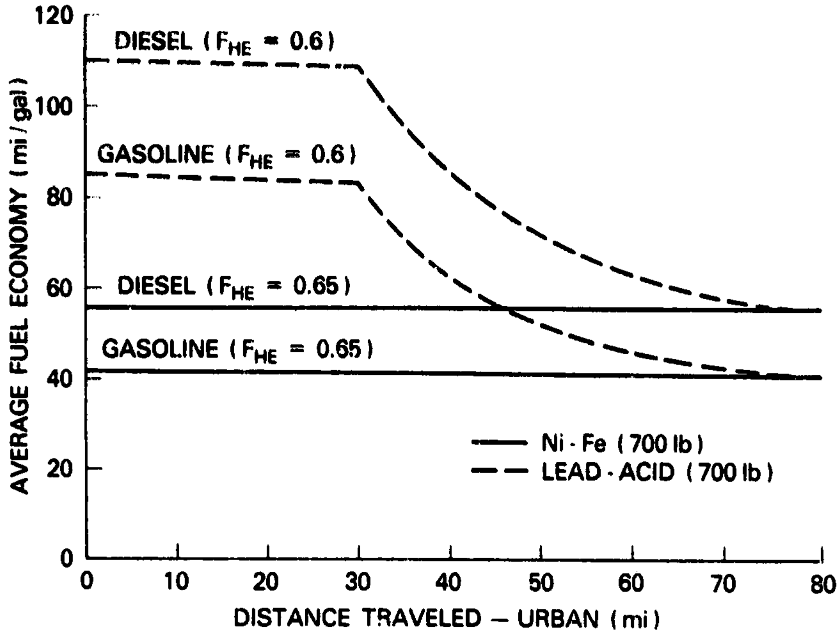


Figure 8-18. Comparison of the Fuel Economy of Hybrid Vehicles Using Ni-Fe and Lead-Acid Batteries

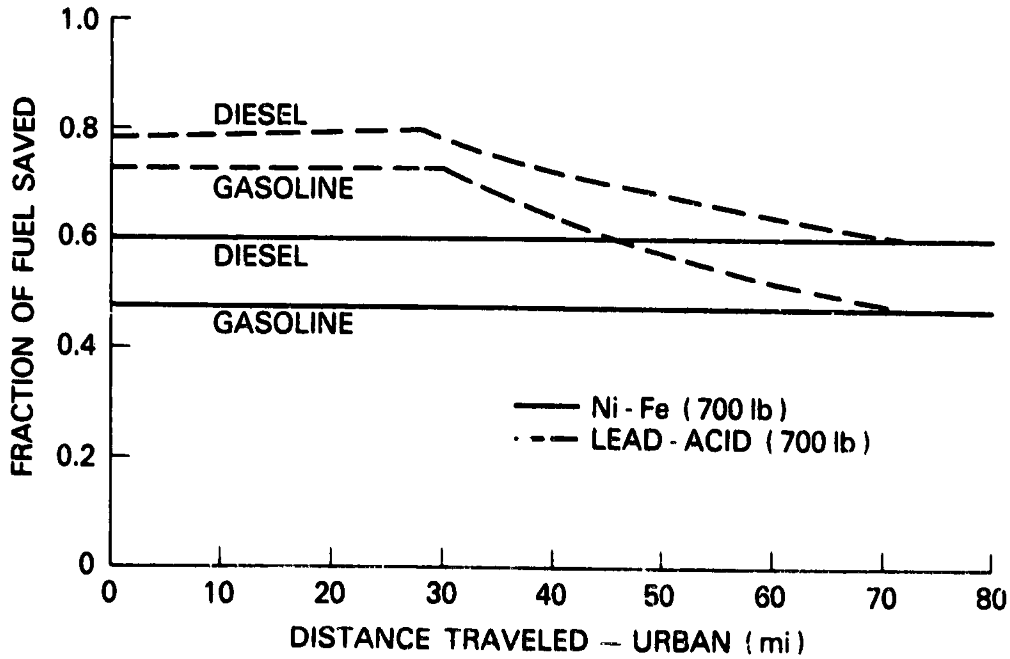


Figure 8-19. Comparison of Fraction of Fuel Saved for the Hybrid Vehicle Using Ni-Fe and Lead-Acid Batteries

Figures 8-14 through 8-16 indicate that Ni-Zn batteries would be very attractive for use in hybrid vehicles. The combination of relatively high energy density and high power density are ideal for the hybrid application. As discussed in Section 3.3, there is considerable uncertainty whether Ni-Zn batteries with sufficient life (at least 500 cycles) can be developed in time for use in the Near-Term Hybrid Vehicle Program. If such batteries are available, they certainly would be prime candidates for use in the present program. Otherwise, lead-acid batteries (700 lb) would be used. The vehicle-simulation results indicate that the lead-acid batteries would yield satisfactory vehicle performance, gasoline savings, and life-cycle cost.

Hybrid vehicle simulation results using Ni-Fe batteries are presented in Figures 8-17 through 8-19. All the results shown are for 700 lb of batteries. Even with the use of 700 lb of Ni-Fe batteries, it was necessary to increase the fraction of the driveline peak power contributed by the heat engine from 60 to 65% in order for the hybrid vehicle to meet the maximum accelerations encountered in the EPA Urban Cycle without the battery voltage falling below that required by the motor. The internal resistance of the Ni-Fe batteries is so high that the electric motor cannot draw the high armature currents required to produce peak power without creating a large voltage drop in the battery. Hence, even when the batteries are near full charge and the vehicle speed is relatively low (less than VMODE), the heat engine must share the load with the electric motor. The result is that the battery is depleted much more slowly using Ni-Fe batteries than using lead-acid and the fuel economy of the hybrid vehicle for the first 30 mi is much less using Ni-Fe and does not show as pronounced a decrease at extended ranges as lead-acid batteries. It is of interest that at about 75 mi the average fuel economy and fraction of fuel saved using Ni-Fe batteries are equal to that using lead-acid batteries. For longer distances a hybrid vehicle using Ni-Fe batteries would show an advantage over one using lead-acid batteries because the former could continue to operate in an electric primary mode, while the latter would have to recharge the batteries from the heat engine. The vehicle simulation results for Ni-Fe batteries indicate that they are not suitable for a hybrid vehicle design whose goal is to save as much fuel as possible in urban driving involving relatively low daily mileage (less than 50 mi). Hence, it was decided not to consider further the use of Ni-Fe batteries in the Near-Term Hybrid Vehicle Program. Lead-acid and Ni-Zn batteries offer a more advantageous combination of energy density and power density than the Ni-Fe and, thus, they are considered the prime battery candidates for the hybrid vehicle application.

The acceleration performance characteristics of the hybrid vehicle using various battery and heat engine combinations are summarized in Table 8-8. The table indicates that the performance of vehicles using lead-acid and Ni-Zn batteries are nearly identical, and that even using 500 lb of Ni-Zn batteries, the battery is not the power-limiting element in the electric drive system.

Table 8-8

ACCELERATION PERFORMANCE CHARACTERISTICS
OF VARIOUS HYBRID POWER TRAIN COMBINATIONS

Engine Type/Power (kW)	Electric Motor Power (kW)	Battery Type/ Weight (lb)	Acceleration Time* (sec) 0 to 50 Km/h	0 to 100 Km/h
Gasoline/55	36	Lead-Acid/700	3.5	12.1
Gasoline/45	36	Lead-Acid/700	4.6	15.1
Gasoline/55	36	Ni-Zn/700	4.1	13.2
Gasoline/55	36	Ni-Zn/500	4.1	13.5
Gasoline/59	32	Ni-Fe/700	5.7	18.6
Diesel/55	36	Lead-Acid/700	3.9	12.5
Diesel/36	36	Lead-Acid/700	4.8	15.8
Diesel/55	36	Ni-Zn/700	3.9	12.5
Diesel/55	36	Ni-Zn/500	3.9	12.5
Diesel/59	32	Ni-Fe/700	5.2	16.7

*Batteries at full charge

The acceleration times for vehicles using Ni-Fe batteries are significantly longer than those using lead-acid and Ni-Zn batteries because the battery limits the power delivered from the electric motor.

The all-electric operation of the hybrid vehicle using 700 lb of lead-acid batteries was calculated for the SAE J227a B schedule and the stabilized portion of the EPA urban cycle. The results for battery energy used per mile (kWh/mi) and range, assuming 100% discharge of the batteries, are given in Table 8-9. Both cycles use about 0.3 kWh/mi and yield a range of 45 to 50 mi. This range seems acceptable for operation of the hybrid vehicle in the all-electric mode in slow city traffic.

Table 8-9

ALL-ELECTRIC OPERATION OF THE HYBRID
VEHICLE IN SLOW CITY TRAFFIC

Cycle	Distance Per Cycle (mi)	Battery Energy Used $\left(\frac{kWh}{mi}\right)$	Range (mi)*
SAE J227a-B	7.50	0.285	54
EPA Urban (Stabilized Segment)	3.84	0.318	43

*Range for 100% discharge of battery; 700 lb lead-acid battery

8.6 CONTROL STRATEGY TRADE-OFF

Considerable work was done developing the various control strategies presently in the HYVEC program. Some of the trade-offs considered are discussed in detail in Section 7. One of the key trade-offs which was evaluated using HYVEC was the effect of on/off engine operation compared with idling the heat engine when it is not the primary drive system. Previous studies[15] have indicated that this is a very important consideration in developing the control strategy for a hybrid vehicle. The effect of engine idling on the fuel economy of the hybrid vehicle in urban driving is shown in Figures 8-20 and 8-21 for gasoline and diesel engines. As expected, the degradation in fuel economy is large due to idling the engine rather than shutting it off when it is not needed to provide power. For the gasoline engine, the fuel economy is reduced by 39% by idling the engine, and for the diesel engine the reduction is 26%. Hence, it is apparent that every effort should be made to shut off the heat engine unless it is needed to provide power or to drive an accessory, such as the air conditioner, during the pull-down period. Engine warmup time should, of course, also be held to a minimum. It is of interest to note that the diesel engine in the idling mode of operation yields about the same fuel economy as the gasoline engine in the on/off operating mode.

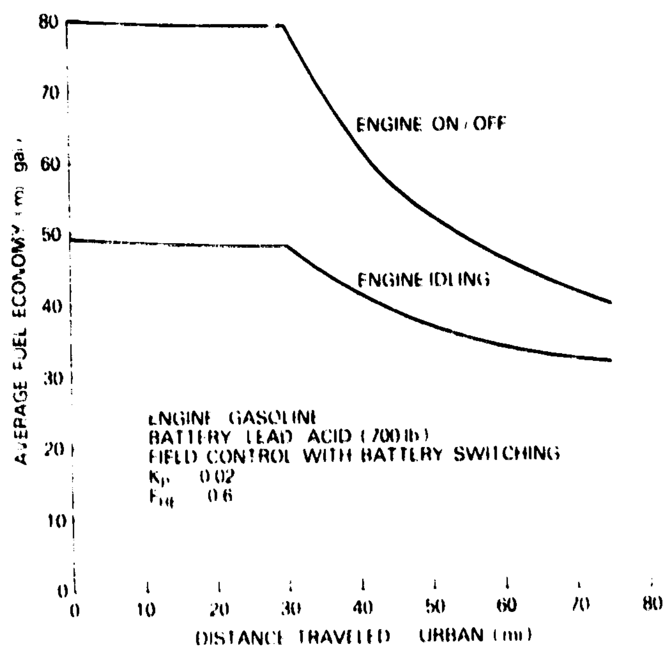


Figure 8-20. The Effect of Engine Idling on Fuel Economy for Urban Driving

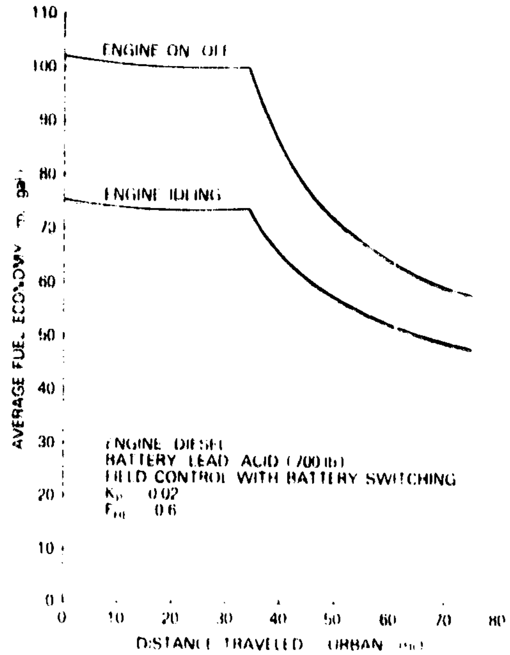


Figure 8-21. Effect of Engine Idling on Fuel Economy in Urban Driving

Section 9

GUIDELINES FOR THE PRELIMINARY DESIGN TASK AND VEHICLE PERFORMANCE AND ENERGY-USE CHARACTERISTICS

Section 9

GUIDELINES FOR THE PRELIMINARY DESIGN TASK AND VEHICLE PERFORMANCE AND ENERGY-USE CHARACTERISTICS

9.1 INTRODUCTION

Based on the Design Trade-off Studies discussed in previous sections, guidelines have been developed for the Preliminary Design Task. These guidelines will be followed in preparing the detailed layouts of the vehicle chassis, body, and power train and writing specifications for the various driveline components, in assessing the microprocessor requirements for implementing the control strategy, and in the final detailed computer simulations of vehicle performance and energy usage (fuel and electricity). The guidelines are discussed in three parts:

- Vehicle layout and chassis
- Power train configuration and components
- Control strategy

9.2 VEHICLE LAYOUT AND CHASSIS

The following are characteristics of the vehicle layout and chassis.

- Curb weight - 3800 lb
- Body style
  - Four-door Hatchback
  - Drag coefficient - 0.40
  - Frontal area - 21.5 ft^2
- Chassis/Power Train arrangement
  - Front-wheel drive
  - Complete power train in front of fire wall
  - Fuel tank under rear seat
- Baseline ICE Vehicle
  - 1979 Chevrolet Malibu

9.3 POWER TRAIN CONFIGURATION/COMPONENTS

The configuration and components of the power train include the following:

- Power train configuration
  - Parallel hybrid
- Heat engine
  - 73 hp
  - Fuel-injected gasoline or turbocharged diesel

- Electric drive system
  - Direct current separately excited motor
  - 18 kW (continuous rating)
  - Field control and battery switching
- Batteries
  - 700 lb lead-acid
  - Or 500 lb Ni-Zn
- Transmission
  - Three or four speed automatically shifted gear box
  - Or steel belt continuously variable transmission
- Power combination (input/output)
  - Single-shaft power combination with fixed ratio between input and output shafts
  - Or power differential with grounded and over-running clutches to lock differential for all-electric and heat engine driving

9.4 CONTROL STRATEGY

The control strategy will include the following:

- On/off engine operation
- Regenerative braking
- Electric motor idling
- Electric drive system primary-battery state of discharge permitting vehicle speed less than VMODE
- Equal sharing of load between motor and engine when both are needed
- Batteries recharged by heat engine in narrow range ($0.7 < S < 0.8$)
- Electric motor dominant in determining shifting logic when it is operating
- Heat engine primary for highway driving when vehicle speed is greater than VMODE
- Electric motor is always used to initiate vehicle motion from rest and in low speed maneuvers (e.g., parking)
- Vehicle operation controlled by a system microprocessor

9.5 VEHICLE PERFORMANCE AND ENERGY-USE CHARACTERISTICS

The performance and energy-use characteristics of the hybrid vehicle (gasoline engine powered) to be designed in the Preliminary Design Task are summarized in this section. The format used to present the vehicle characteristics is that used by the Jet Propulsion Laboratory in Exhibit I of Request for Proposal No. HI-2-8275, February, 1977. The values given in the following tables, Vehicle Performance Specifications and Energy Consumption Measures, are based on HYVEC program calculations for the power train and control strategy described in Sections 9.3 and 9.4, respectively.

Table 9-1

VEHICLE PERFORMANCE SPECIFICATIONS

P1	Nonrefueled Range	
	P1.1 [illegible] (Gasoline 10 gal Tank)	[illegible] Km
	P1.2 [illegible]*	120 Km, 4[illegible] Km
	P1.3 [illegible] (Electricity Only)	80 Km
P2	Cruise Speed	[illegible] Km/h
P3	Maximum Speed	
	P3.1 Maximum Speed	[illegible] Km/h
	P3.2 Length of Time Maximum Speed Can Be Maintained on Level Road	1 min.
P4	Accelerations	
	P4.1 0-50 Km/h (0-30 mph)	4.0 s
	P4.2 0-90 Km/h (0-56 mph)	11.0 s
	P4.3 40-90 Km/h (25-56 mph)	[illegible].8 s

P5 Gradability	Grade	Speed	Distance
P5.1	3	[illegible] Km/h	Km (Unlimited)
P5.2	[illegible]	[illegible] Km/h	Km (Unlimited)
P5.3	8	[illegible] Km/h	Km (Unlimited)
P5.4	1[illegible]	1[illegible] Km/h	Km (Unlimited)
P5.5 Maximum Grade	[illegible]		

P6	Payload Capacity (Incl. Passengers)	[illegible] Kg
P7	Cargo Capacity	0.[illegible] m^3
P8	Consumer Costs	
	P8.1 Consumer Purchase Price (19[illegible] $)	[illegible]000
	P8.2 Consumer Life Cycle Cost (19[illegible] $)	0.11 $/Km
P9	Emissions - Federal Test Procedure** (Gasoline Engine)	
	P9.1 Hydrocarbons (HC)	[illegible].06 gm/Km, 0.12 gm/Km
	P9.2 Carbon Monoxide (CO)	0.40 gm/Km, 0.[illegible] gm/Km
	P9.3 Nitrogen Oxides (NO_x)	0.40 gm/Km, 0.64 gm/Km

*First number is the range at which the batteries must be recharged from the heat engine; second number is range at which the 40 liter gasoline tank is empty

[illegible]On heat engine alone

**The first number corresponds to first 50 mi, second to 1[illegible]0 mi

Table 9-2

ENERGY CONSUMPTION MEASURES

E1	Annual petroleum fuel energy consumption per vehicle compared to reference vehicle over contractor-developed mission[a]	30,000 MJ SAVED[b]
E2	Annual total energy consumption[c] per vehicle compared to reference vehicle over contractor-developed mission[a]	6,000 MJ SAVED[b]
E3	Potential annual fleet petroleum fuel energy savings compared to reference vehicle over contractor-developed mission[c]	30×10^9 MJ
E4	Potential annual fleet total energy consumption[c] compared to reference vehicle over contractor-developed mission[d]	6×10^9 MJ SAVED[b]
E5	Average energy consumption[c] over maximum nonrefueled range	
	E5.1 FHDC (gasoline only)	2.53 MJ/km (32 mpg)
	E5.2 FUDC[e]	2.86 MJ/km, 3.1 MJ/km, 3.6 MJ/km
	E5.3 J227a (B) (electricity only)	2.45 MJ/km
E6	Average petroleum fuel energy consumption over maximum nonrefueled range	
	E6.1 FHDC	2.53 MJ/km (32 mpg)
	E6.2 FUDC[e]	1.01 MJ/km (80 mpg), 1.93 MJ/km (42 mpg), 3.2 MJ/km (25 mpg)
	E6.3 J227a (B)	0 MJ
E7	Total energy consumed[c] versus distance traveled starting with full charge and full tank over the following cycles	
	E7.1 FHDC	2.53 MJ/km (Not a Function of Distance)
	E7.2 FUDC	(See Figure 8-8)
	E7.3 J227a (B)	2.45 MJ/km (Not a Function of Distance)
E8	Petroleum fuel energy consumed versus distance traveled starting with full charge and full tank over the following cycles[f]	
	E8.1 FHDC	2.53 MJ/km (Not a Function of Distance)
	E8.2 FUDC	(See Figure 8-1)
	E8.3 J227a (B)	0 MJ/km (Not a Function of Distance)

1 MJ = 0.278 kWh = 948 Btu = .00758 gal gasoline

10^9 MJ/yr = [illegible] barrels crude oil/day

(a) Mission is 11,852 mi/yr; 65% EPA urban cycle, 35% EPA highway cycle

(b) The annual fuel and energy usages of the Reference ICE Vehicle (1985 model) are 456 gallons of gasoline and 60,158 MJ. A fleet of one million Reference vehicles would use 60×10^9 MJ.

(c) Includes energy needed to generate the electricity at the power plant (35% efficiency)

(d) For one million hybrid vehicles replacing one million Reference Vehicles

(e) The first number corresponds to the first 50 km; the second number to 120 km; the third number to 425 km, at which the gasoline tank is empty

(f) Does not include petroleum consumption resulting from generation of wall plug electricity used by the vehicle

Section 10

SUMMARY AND MAJOR FINDINGS

Section 10

SUMMARY AND MAJOR FINDINGS

10.1 SUMMARY

Design trade-off studies were made to determine the relative attractiveness of various hybrid/electric power train configurations and electrical and mechanical drive-line components. Initial screening of the candidate configurations and components was done using a vehicle synthesis computer program. The initial screening was concerned primarily with total vehicle weight and economic factors and identified the hybrid power train combinations which warranted detailed evaluation over various driving cycles. This was done using a second-by-second vehicle simulation program which permitted the calculations of fuel economy, electricity usage, and emissions as a function of distance traveled in urban and highway driving. Vehicle layout studies were also made to evaluate various power train arrangement possibilities in terms of their effect on vehicle handling, safety, serviceability, and passenger comfort. Based on the design trade-off studies, the power train layout and components were selected for the Preliminary Design Task of the Near-Term Hybrid Vehicle Program.

10.2 MAJOR FINDINGS

The major findings from the design trade-off studies are:

1. The parallel configuration with a 60/40 split between peak power of the heat engine and electric drive systems was near-optimum from the standpoints of vehicle weight, ownership cost, and energy usage (fuel and electricity).
2. Based primarily on economic considerations, a dc electric drive system utilizing a separately excited motor with field control and battery switching was selected for the Near-Term Hybrid Vehicle.
3. The prime heat engine candidates are a fuel-injected gasoline engine and a turbocharged diesel. Both engines are 1.6 ℓ in displacement and develop about 70 hp. The diesel engine yielded 25 to 30% better fuel economy in the hybrid application than the gasoline engine, but technology does not currently exist to reduce the NO_x and particulate emissions of the diesel to levels being considered by the Environmental Protection Agency for 1985. The diesel also has possible cold-starting problems when used in an on/off mode.
4. A complex control strategy involving integrated power sharing between the heat engine and the electric drive systems is required for the hybrid vehicle to have acceleration performance equivalent to a conventional ICE vehicle and at the same time high fuel economy and

acceptable electric range. Implementation of the control strategy developed in the computer simulations will require the use of microprocessors in the hybrid vehicle control system.

5. The hybrid vehicle simulations showed that 700 lb of ISOA lead-acid batteries yielded satisfactory electric range and vehicle acceleration performance. The Ni-Zn batteries were found to be the most attractive for the hybrid application, but there is considerable uncertainty concerning the cycle lifetime and cost of Ni-Zn batteries in the 1982 to 1985 time period.

6. The vehicle layout studies showed that the complete hybrid power train including the lead-acid batteries could be packaged in the engine compartment of the 1979 Chevrolet Malibu without any intrusion into the passenger compartment.

7. The initial selling price (in 1978 dollars) of the hybrid vehicle was calculated to be about $7000 compared with $5700 for a conventional ICE vehicle of the same performance and passenger-carrying capacity. The ownership (life cycle) cost of the hybrid was calculated to be 17.8¢/mi compared with 18.5¢/mi for the Reference Vehicle for energy costs of $1.00/gal for gasoline and 4.2¢/kWh for electricity. The lifetime of the hybrid vehicle was taken to be 12 yrs compared with 10 yrs for the conventional ICE vehicle.

8. Detailed hybrid vehicle simulations showed that for the first 30 mi (the electric range of the vehicle) in urban driving, the fuel economy was 80 mpg using a gasoline engine and 100 mpg using a diesel engine. Over the first 75 mi the average fuel economy of the hybrid was 42 mpg for the gasoline engine and 55 mpg using the diesel engine. The highway fuel economy of the hybrid vehicle is slightly better than that of the Reference ICE Vehicle. In urban driving the hybrid would save about 75% of the fuel used by the conventional vehicle and in combined urban/highway driving the fuel saving is about 50%.

Section 11

REFERENCES

Section 11

REFERENCES

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